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SUMMARY

This report describes a computer program, TLIFE, which simulates the life, dynamic capacity and reliability of aircraft transmissions. The program is written in Fortran 77 and has been executed both in the personal computer DOS environment and on UNIX work stations. Primary interaction with the program is through ASCII disc files, with input provided by a ".in" file and output written to a ".out" file with the same prefix. Some user keyboard input is required and the analysis output is written to the screen during execution. The source code file is 5,101 lines long and 167 kB in size, while the executable DOS ".exe" file is 157 kB in size. Analysis of a single transmission occurs in a few seconds on a 286 PC equipped with a numerical co-processor.

A variety of transmissions may be analyzed including spur reductions, helical reductions, spiral-bevel reductions and combinations of these. The basic spur and helical reductions include single-mesh, compound and parallel-path gear trains. Additional spur-gear drives which can be analyzed are the star and planetary reductions possible with reverted and single-plane configurations. The final gear may be an internal (ring) or external gear, except in the case of the single-plane reduction which must have an internal final gear. A variety of straddle and overhung bearing configurations on the input, intermediate and output shafts are possible. The basic spiral-bevel reductions include single-input and dual-input drives. Complete transmissions composed of one or more of the above reductions can be analyzed by the program with the limitation that the dual spiral-bevel reduction can appear only once.

The analysis may be performed in the SI metric or the English Inch system. The program prompts the user for a data file prefix name, takes the input from an ASCII file with that prefix and a '.in' extension and writes the output to a second ASCII file with the same prefix and a '.out' extension. The input file includes: the transmission configuration, input torque and speed, and a description of the component gears and bearings and their locations.

To report the analysis, the program output file describes the overall transmission and each constituent unit transmission, its components, locations, their capacities and loads. It also lists the dynamic capacity and ninety-percent reliability and mean life of: each component, each unit transmission and the overall transmission as a system. The overall component mean life is given to describe the overhaul frequency when only failed components are repaired.

The reliability and life analysis is based on the two-parameter Weibull distribution lives of the transmission gears and bearings. The system dynamic capacity, ninety-percent reliability life and mean life of the system are based on a strict series probability model for the transmission reliability. A component or system's dynamic capacity is represented by the transmission output torque which can be applied for one-million output shaft rotations with a probability of survival for that component or system of ninety percent. Ninety-percent reliability life is the life of the component or system in million output rotations or hours at which ninety percent of identical components or systems would survive under the given loading conditions.

This report describes the use of the program, its input and output files and some theory behind its analysis. Examples are presented to illustrate the information available for single-element and series transmissions.

INTRODUCTION

There is a need for new transmissions that have longer service lives and yet are lighter and occupy less space than present transmissions [1]. An important property is the service time between overhauls. Since the lives of aircraft transmissions are high, the testing of different transmissions is a long process which is both time consuming and expensive. The time involved in selecting an optimal transmission by experimental testing is extensive, since many long tests are required to obtain accurate results. In cases such as this, the computer comes to the aid of the designer [2,3].

Computer programs are available for the life analysis of different bearings using the Lundberg-Palmgren fatigue life model [4]. This theory has also been applied effectively for the analysis of fatigue lives of spur and helical gears assuming surface pitting as the eventual mode of failure [5, 6]. The basis of the reliability analysis is the two-parameter Weibull distribution.

There are also analysis programs for life and dynamic capacity at a given reliability for planetary and bevel gear transmissions [7,8,9] which include a number of reductions presently in use and for a number of parallel shaft reductions [10].

These simulations have limitations in terms of the configurations they can analyze. However, the parallel shaft reduction program (PSHAFT) [10] took on a modular form to allow the addition of new configurations. The current simulation (TLIFE) adds significantly to the number and complexity of transmissions which can be analyzed. Helical gear transmissions and single and dual-input spiral-bevel transmissions have

been added to the basic transmission set and a method of combining basic transmissions to simulate more complex systems has been devised. This saves the effort and time involved in developing code to simulate a new transmission which is composed of the same unit transmissions in a different order. Also, the modular construction of the program simplifies adding a new unit configuration to the program.

This report describes the use and methods of the program for life and reliability analysis of aircraft transmissions. The program has been written in ANSI standard FORTRAN 77 and runs both on the personal computer in the DOS environment and on the main frame computer in the UNIX environment.

The life and reliability analysis in this simulation is based on the assumption that the transmissions are well lubricated and the gears well designed and of high quality. Failure by tooth breakage and tip scoring are avoided thereby and surface pitting is the only mode of failure [5,11,12].

The input file includes the transmission configuration, specification of input torque and input speed for the first transmission and the locations, types and nominal capacities of the gears and bearings for all unit transmissions of which the overall transmission is composed from input to output.

The output file first gives the description of transmission configuration, transmission characteristics, the input data for the gears and bearings followed by a report of the loads and capacities, and a summary of the dynamic capacities of the different components in units of output torque, ninety-percent reliability lives in million output rotations

and hours and the mean life in hours for each of the component unit transmissions.

Concluding the output are the transmission mean life and overall mean component life in hours. The transmission life predicts the mean life between service overhauls with full transmission replacement. The overall component mean life predicts the mean time between service overhauls for maintenance by failed component replacement only. Also, when analyzing series transmissions, the program gives the life of the overall transmission based on the lives of individual unit transmissions. This is appropriate when the different unit transmissions are considered as separate units.

PROGRAM CAPABILITY

The program analyzes a transmission for its service life as measured by the ninety-percent reliability lives of its components and system and by the mean life of the components, system and the sum of the components. The transmission may be any one of a list of basic transmissions or may be a series of these unit transmissions.

Basic Transmissions

Eleven basic transmission configurations comprise the units of which transmission systems can be composed for analysis. These unit reductions, shown schematically in Figures 1, 2 and 3, are:

1. The single-mesh spur-gear reduction,
2. The single-mesh helical-gear reduction,
3. The compound spur-gear reduction,
4. The compound helical-gear reduction,
5. The parallel spur-gear reduction,
6. The parallel helical-gear reduction,
7. The reverted spur-gear reduction,
8. The reverted helical-gear reduction,
9. The single-plane spur-gear reduction,
10. The spiral-bevel reduction, and
11. The dual spiral-bevel reduction.

The single-mesh spur reduction is shown in Figure 4. This unit has two spur gears mounted on the input and output shafts supported by two bearings each. These gears can be supported either in straddle or in overhung configurations, as shown in Figure 5. The locations of the bearings are specified by the distances A and B which are measured from the

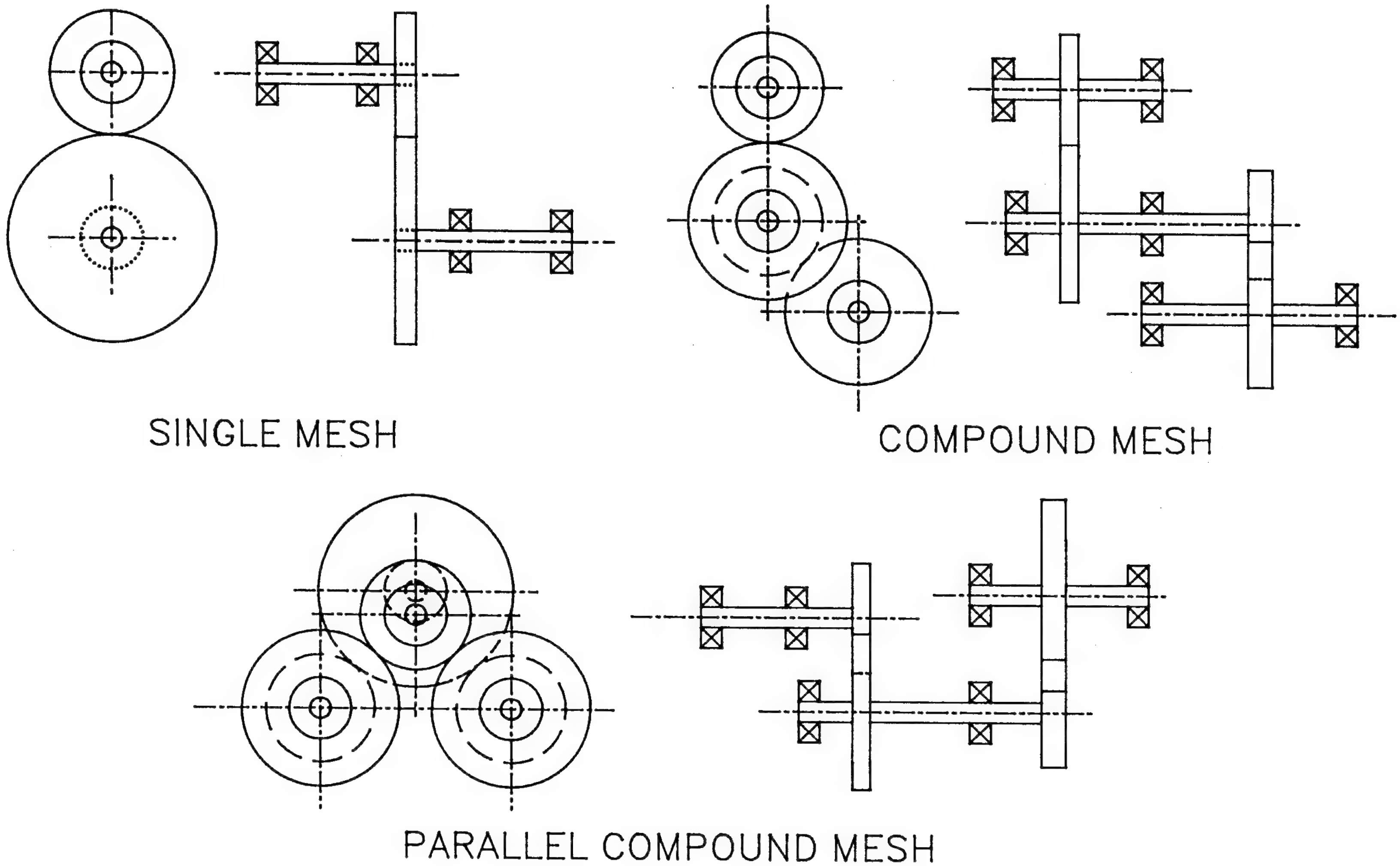
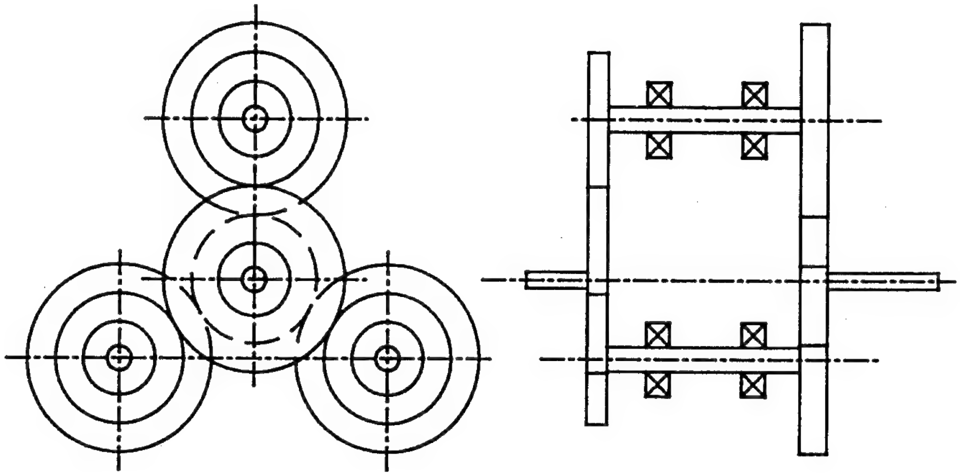
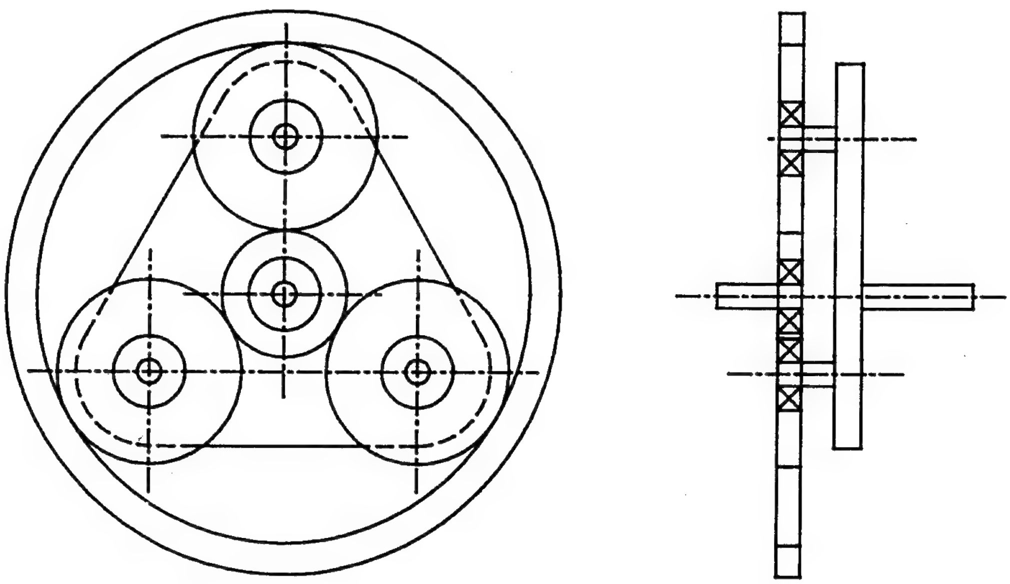


FIGURE 1 — SPUR AND HELICAL UNIT REDUCTION CONFIGURATIONS

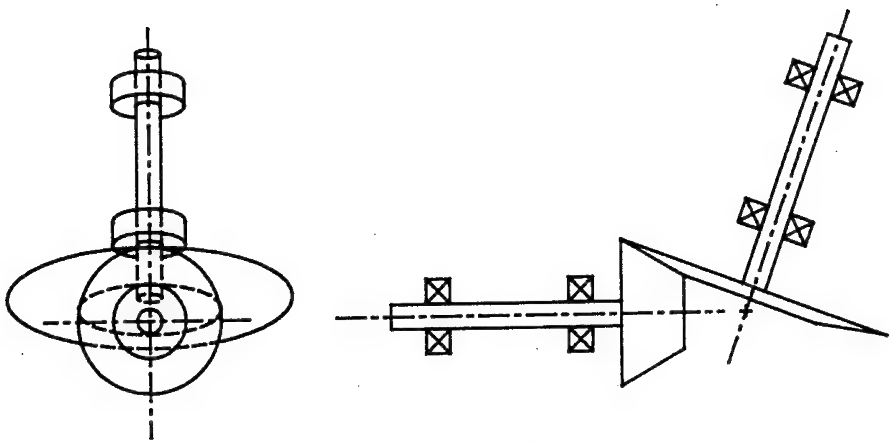


REVERTED SPUR AND HELICAL

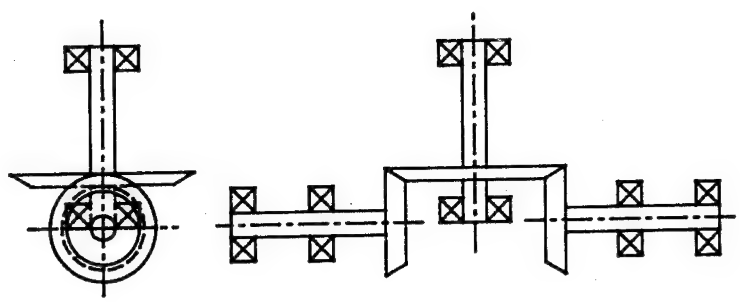


SINGLE PLANE SPUR

FIGURE 2 – STAR AND PLANETARY UNIT REDUCTION CONFIGURATIONS

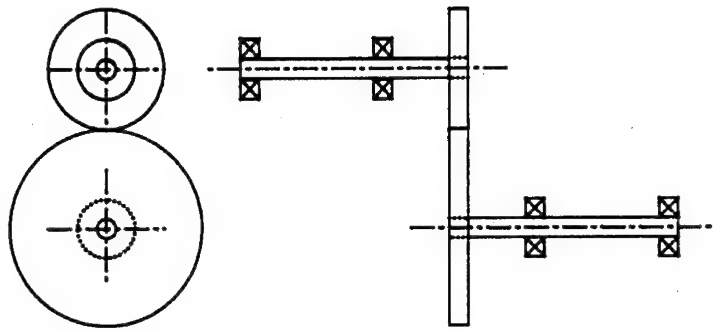


SPIRAL BEVEL

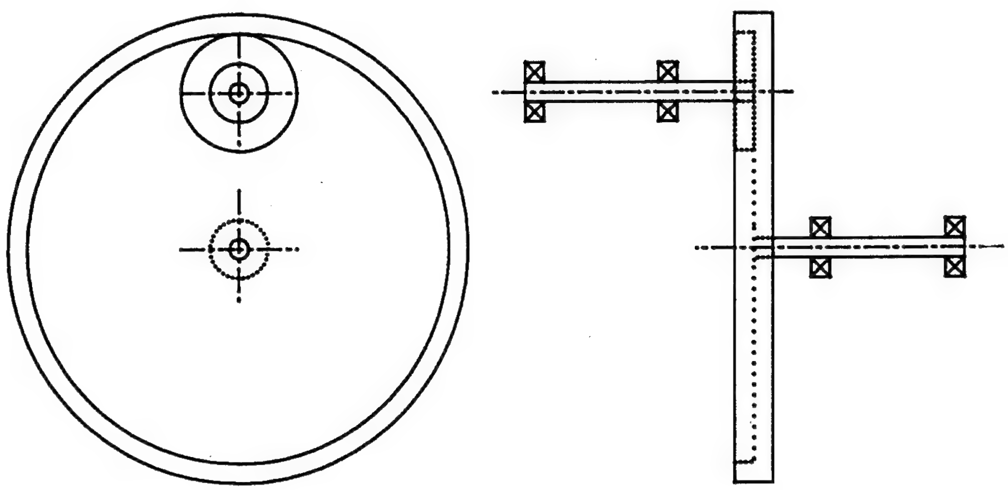


DUAL SPIRAL BEVEL

FIGURE 3 – SPIRAL BEVEL UNIT REDUCTION CONFIGURATIONS

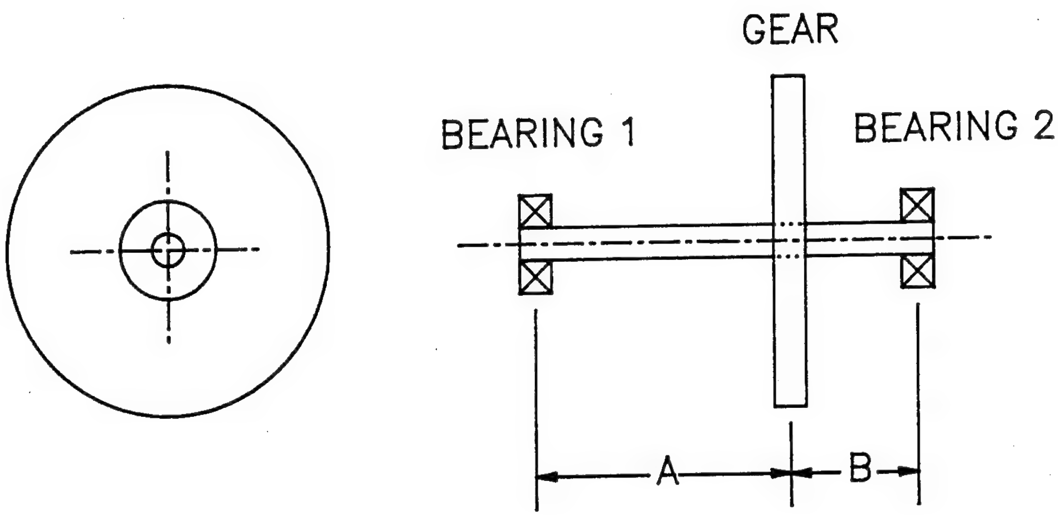


EXTERNAL OUTPUT GEAR

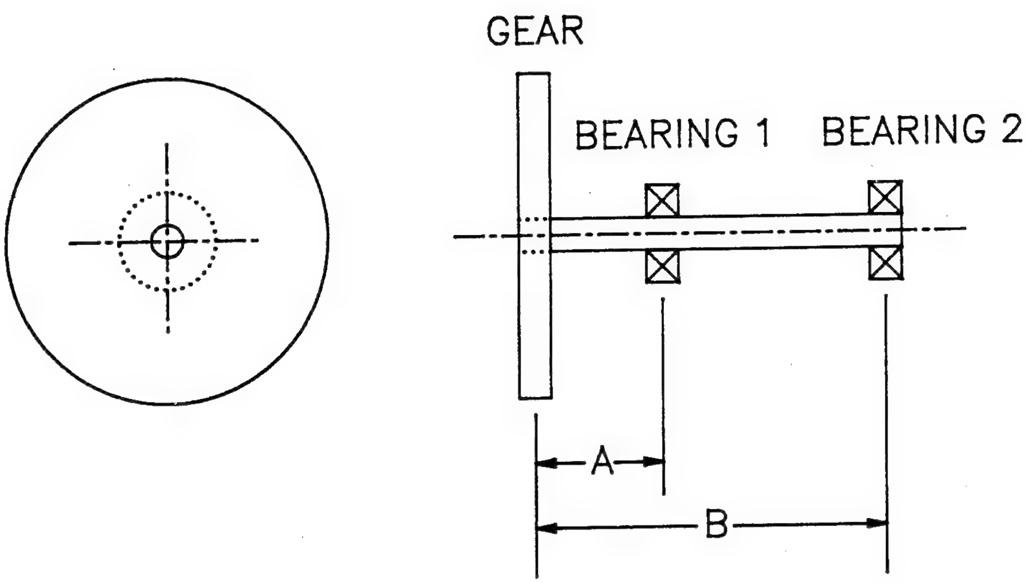


INTERNAL OUTPUT GEAR

FIGURE 4 – SINGLE MESH SPUR AND HELICAL REDUCTIONS



STRADDLE MOUNTING



OVERHUNG MOUNTING

FIGURE 5 – INPUT AND OUTPUT SHAFT MOUNTINGS

gear center to the bearing center. The bearings can be single or double-row ball or cylindrical roller bearings or double-row tapered-roller bearings. As the gears in this reduction are spur gears, it is assumed that there are no thrust loads on the bearings. The output gear may be either an external gear or an internal ring gear as shown in Figure 4.

The single-mesh helical reduction appears schematically the same as its spur-gear counterpart shown in Figure 4. A major difference from the spur reduction is the presence of axial thrust loads on the shafts for helical gears which are not herringbone. On each of the support shafts, at least one of the bearings must be able to support an axial load. The analysis assumes that one bearing takes the thrust load, or that both bearings share the thrust load equally, as desired by the user. The bearings taking the thrust load must be ball bearings or tapered roller bearings as cylindrical roller bearings cannot take thrust load. This unit has configurations which match those of the first unit.

The compound-mesh spur reduction has four spur gears and six bearings as shown in Figure 6. The input and output gears are mounted on shafts, supported by two bearings each and the intermediate shaft, supported on two bearings, carries two intermediate gears. The input and the intermediate gears are external gears and the output gear may be an external gear, or an internal ring gear. The three shafts, supporting the four gears, may be either in one plane, where the input and the output shafts could be collinear, or in different planes, where the input and output shafts are not collinear. The input and output gears may be supported in the same two configurations the single-mesh reduction has, whereas the intermediate

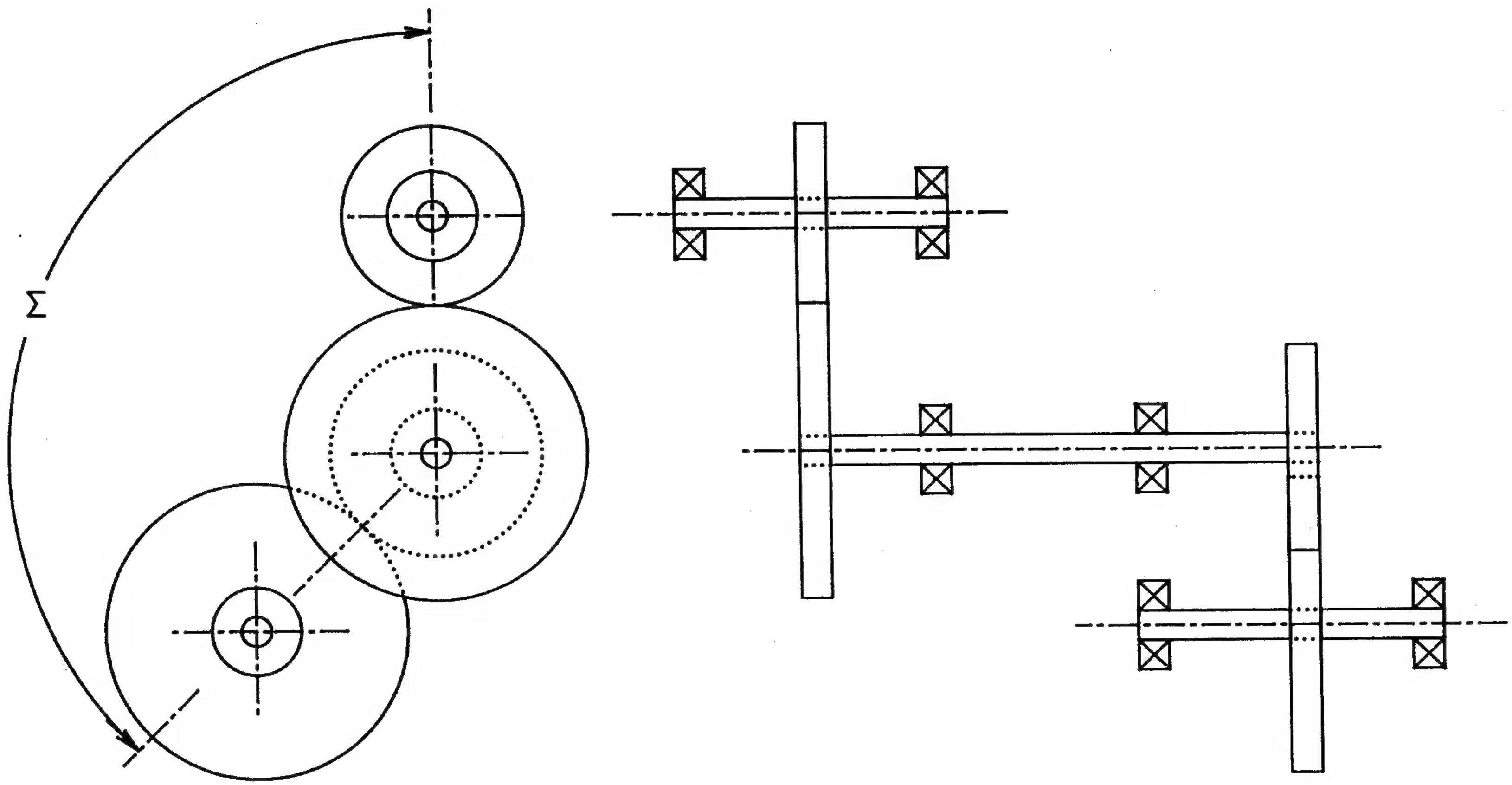


FIGURE 6 — COMPOUND SPUR AND HELICAL REDUCTIONS

gears may be supported in any of the four ways shown schematically in Figure 7, which are:

a) Double straddle,

b) Double overhung,

c) Output gear overhung, and

d) Input gear overhung.

In the double straddle configuration, both intermediate gears are supported between the two bearings. In the double overhung configuration both intermediate gears are supported such that the two bearings are between them. In the output gear overhung configuration, the two bearings straddle the intermediate gear on the input side and are inside the intermediate gear on the output side. In the input gear overhung configuration, the two bearings straddle the intermediate gear on the output side and are inside the intermediate gear on the input side. The distances C, D and E, which locate these bearings, are shown in Figure 7. The distances C and E give the distances of the two bearings from the gears and the distance D gives the distance between the two gears.

Shown in Figure 6 is the shaft angle, Σ, which is measured from the input shaft and intermediate shaft center-line to the intermediate shaft and output shaft center-line, counter-clockwise about the intermediate shaft axis looked at from the input shaft. This angle gives the orientation of the input and output shafts relative to the intermediate shaft. This transmission configuration has a single load path as does the single-mesh reduction.

The compound helical gear reduction has the same schematic representation and configuration possibilities as its spur-gear counterpart

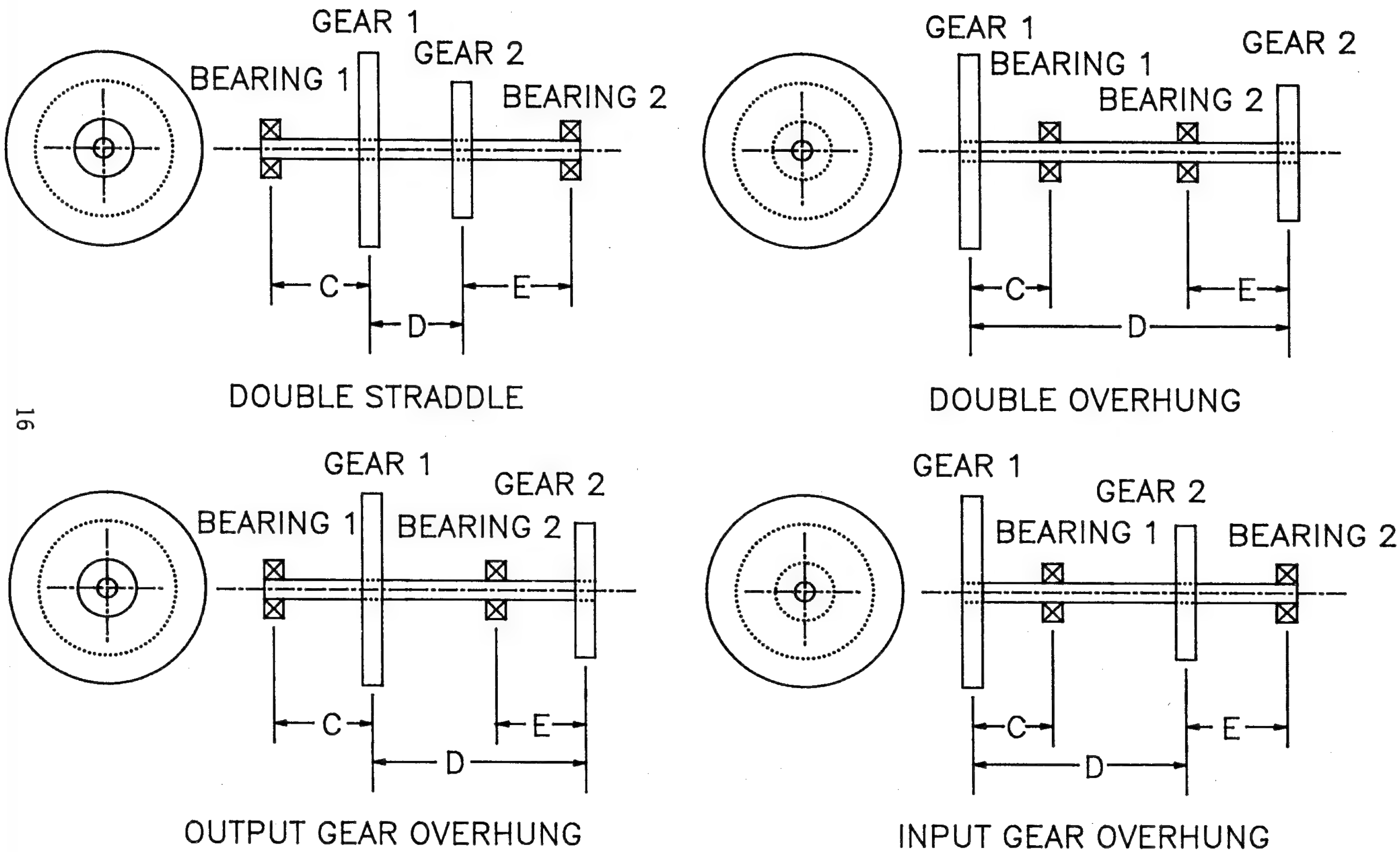
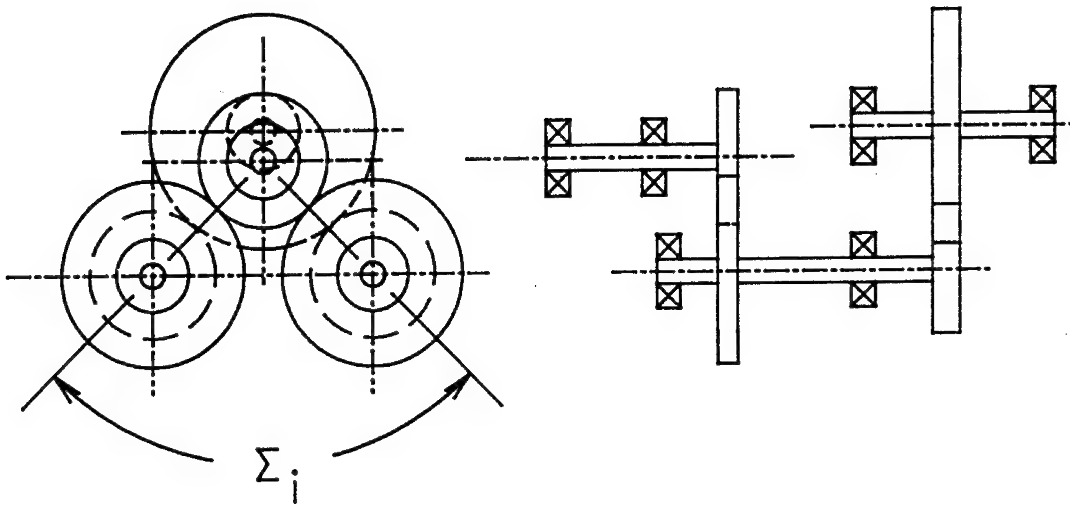


FIGURE 7 — INTERMEDIATE SHAFT MOUNTINGS

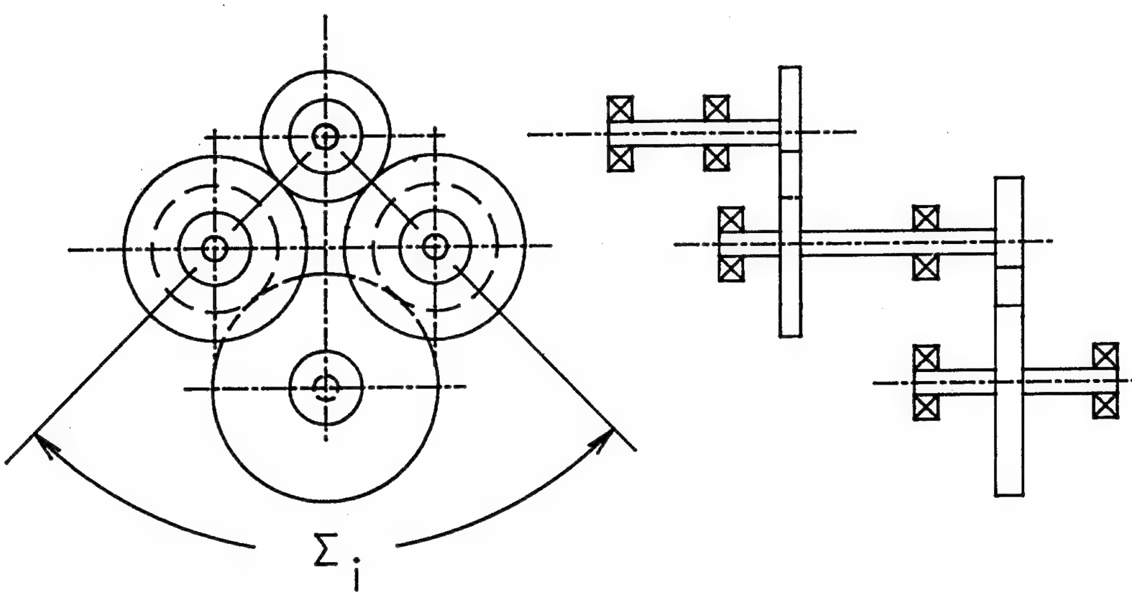
shown in Figure 6. As with the single reduction, a major difference from the spur reduction, for helical gears which are not herringbone, is the presence of axial thrust loads on the shafts. On each of the support shafts, at least one of the bearings must be able to support an axial load.

The parallel compound spur reduction is shown schematically in Figure 8, with its six spur gears and eight bearings. Two possible locations of the output shaft are shown: 1) above the intermediate shafts with the output shaft on the same side of these shafts as the input shaft, and 2) below the intermediate shafts with the output shaft on the opposite side of these shafts as the input shaft. The input and output gears are mounted on shafts supported by two bearings each, and the four intermediate gears are mounted on the two identical intermediate shafts supported by two bearings each. It is assumed that the sub-assemblies of the two intermediate shafts are identical in all respects. The output gear can be an internal or an external gear. For this configuration, the input shaft angle, Σ_i, is the angle that the center distances of the two intermediate shafts with the input shaft make with each other, measured about the input shaft center as shown in Figure 8. The gear mounting configurations for the input, output and the intermediate gears have the same options as the simple compound reduction has, which are shown in Figures 5 and 7.

The parallel compound helical gear reduction has the same schematic representation and configuration possibilities as its spur-gear counterpart shown in Figure 8. As with the single and compound reductions, a major difference from the spur reduction is the presence of axial thrust loads on the shafts. On each of the support shafts, at least one of the bearings must be able to support an axial load unless the gears are herringbone.



OUTPUT SHAFT ABOVE INTERMEDIATE SHAFTS



OUTPUT SHAFT BELOW INTERMEDIATE SHAFTS

FIGURE 8 — PARALLEL COMPOUND
SPUR AND HELICAL REDUCTIONS

The reverted spur reduction, shown schematically in Figure 9, has an input spur gear, a final spur gear, which may be an external gear or an internal ring gear, and at least two identical intermediate shaft assemblies with two spur gears and two bearings symmetrically spaced about the common input gear. The intermediate shaft sub-assemblies may have any of the four configurations shown in Figure 7, with the input and output gear shafts collinear. To complete the geometry, the shaft angle, Σ, is measured between any two adjacent intermediate shaft assemblies as shown in Figure 9. As all the intermediate sub-assemblies are identical and symmetrically spaced about the input gear, the gear loads coming on the input and output shaft bearings are nullified and are not active in the analysis. The output of the reduction unit may be either the final gear, with the arm holding the intermediate shaft sub-assemblies fixed, or the arm with the final gear fixed. Taking the output from the arm makes this configuration work as a reverted planetary reduction.

The reverted helical gear reduction has the same schematic representation and configuration possibilities as its spur-gear counterpart shown in Figure 9. As with the other helical reductions, a major difference from the spur reduction for non-herringbone gears is the presence of axial thrust loads on the shafts. On each of the support shafts, at least one of the bearings must be able to support an axial load.

The single-plane reduction, shown in Figure 10, has an input sun gear and a final ring gear in mesh with a number of planets which are symmetrically placed about the sun and ring gears. This makes the input and output shafts collinear. The planet gears may be un-stepped or stepped, with one size of gears meshing with the sun gear and another size

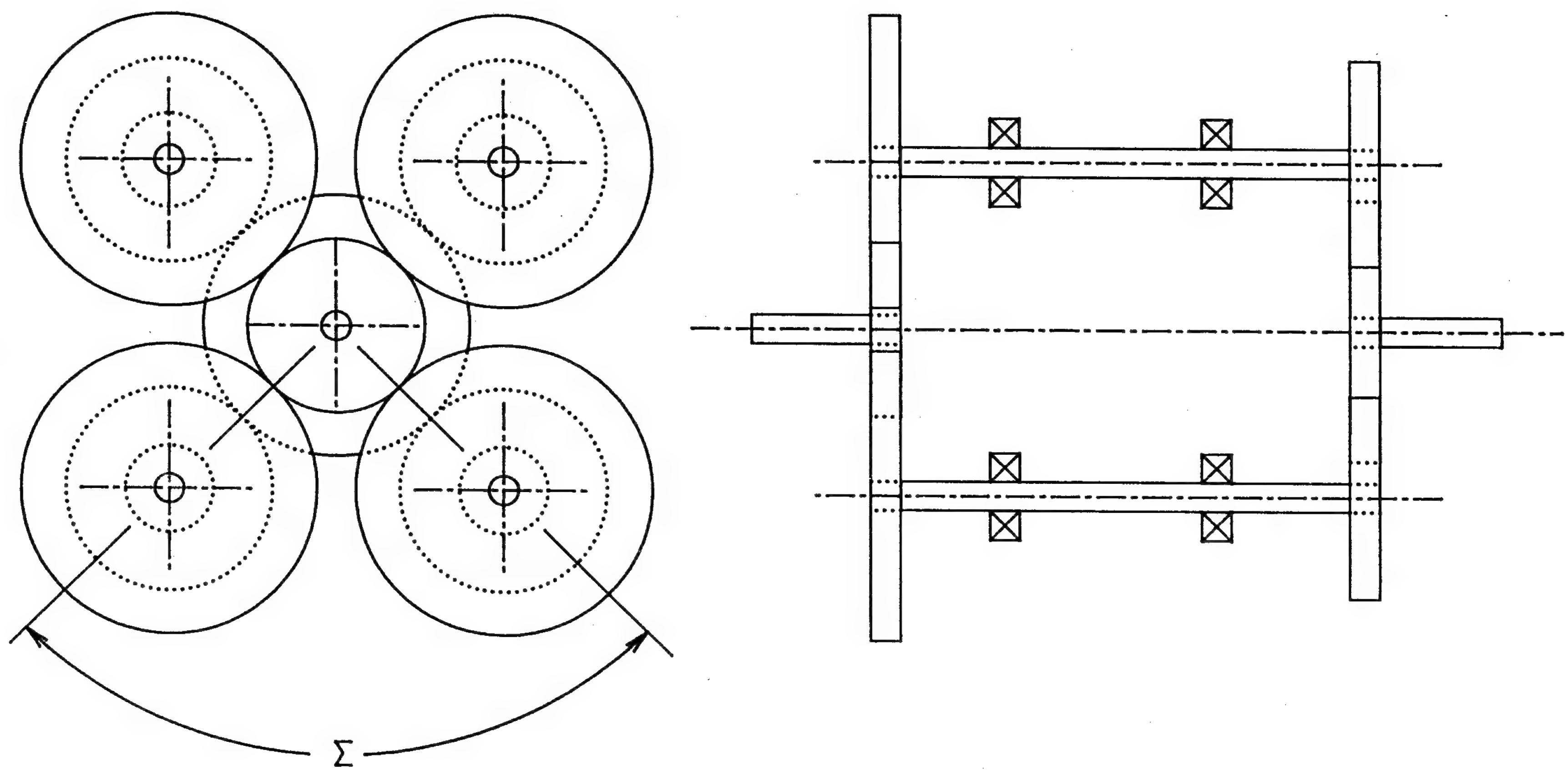


FIGURE 9 — REVERTED SPUR AND HELICAL REDUCTIONS

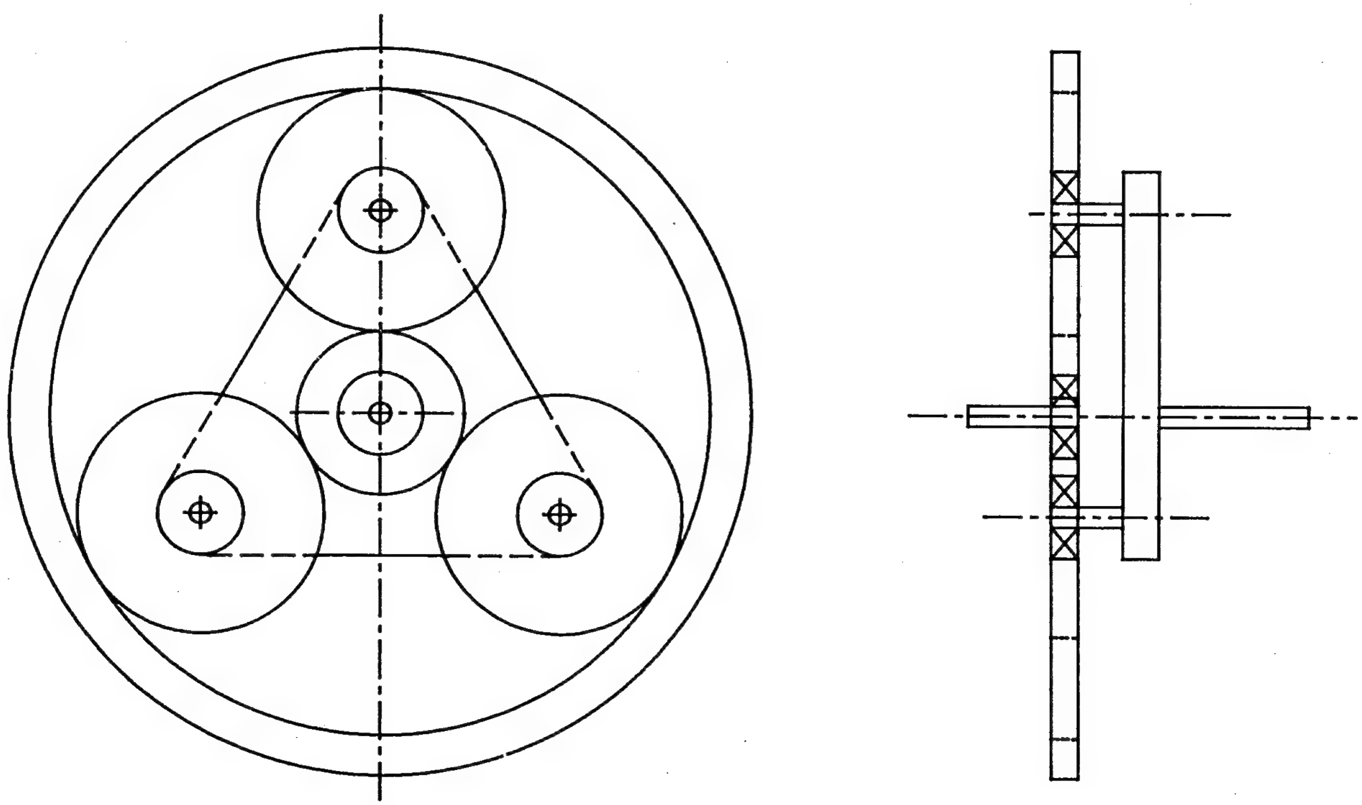


FIGURE 10 — SINGLE PLANE STAR AND PLANETARY REDUCTIONS

of concentric gears meshing with the ring gear. For the stepped planets, one set of planet gears must be split and straddle the other to keep the forces in a single plane. The loads on the planet gears are carried by single, in-plane bearings, and are fixed relative to the inner races. The output of the reduction unit may be either the final gear, with the arm holding the planets fixed, or the arm with the ring gear fixed, providing the output as a planetary reduction.

The spiral-bevel reduction, shown in Figure 11, has two spiral-bevel gears mounted on the input and output shafts, supported by two bearings each. Some of the geometry which is required for the analysis is shown in Figure 11. These gears may be supported either in straddle or overhung configurations as shown in Figure 12, with the gears overhung from the back side. The locations of the bearings are specified by the distances A and B, measured from the gear center to the bearing center. The output gear may be either an external or an internal gear as shown in Figure 13. The analysis assumes that one of the two bearings takes the thrust load, or that both the bearings share the thrust load equally, as desired by the user. The bearings taking the thrust load must be ball bearings or tapered roller bearings, as cylindrical roller bearings cannot take thrust load.

The dual spiral-bevel reduction, the last unit reduction analyzed, is shown in Figure 14. This has three spiral-bevel gears and six bearings. The two input pinions are assumed to be identical in all respects. Each gear is supported by two bearings each. The gears may be supported in the same straddle or overhung configurations as shown in Figure 12. The output gear may be either an external gear or an internal ring gear as shown in Figure 15. Figure 16 shows the location of the two input pinions with

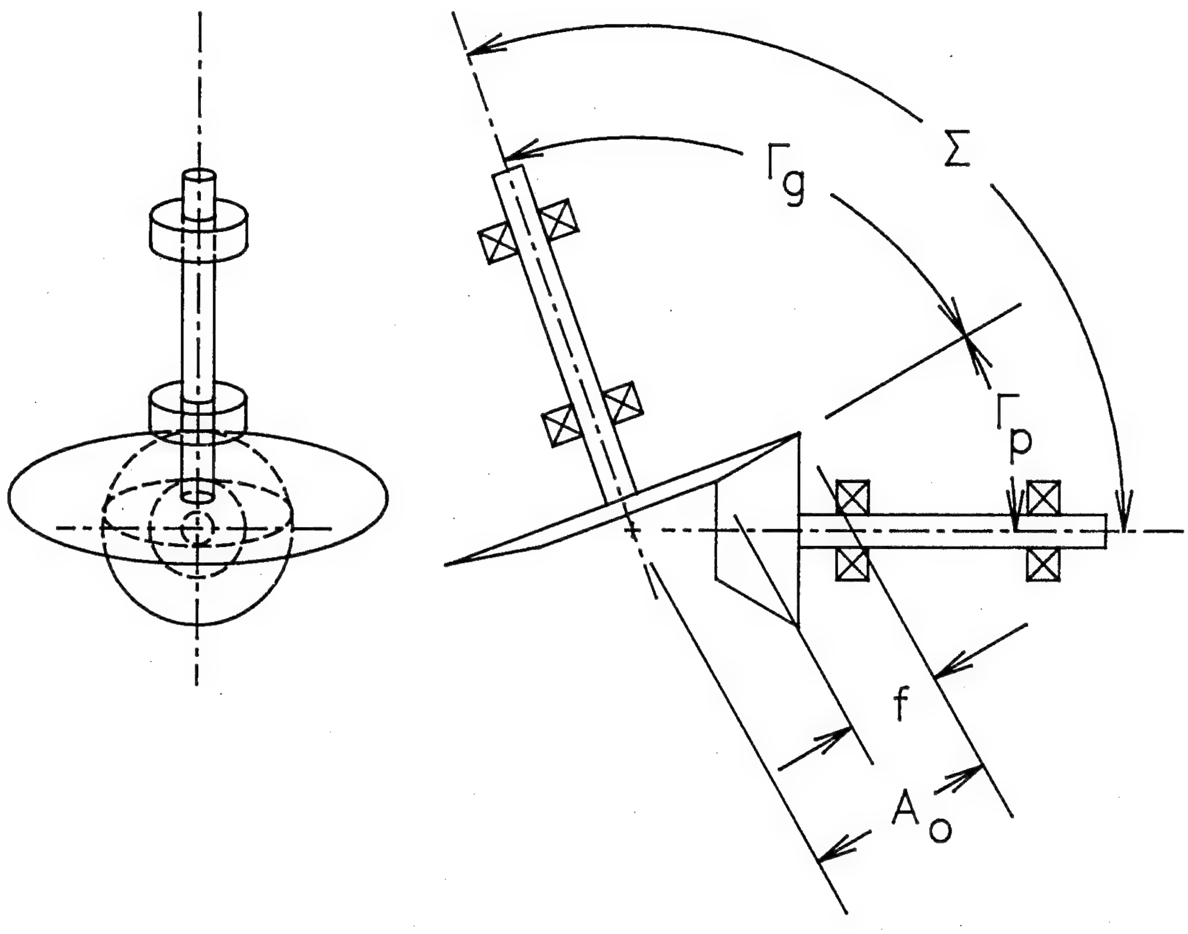
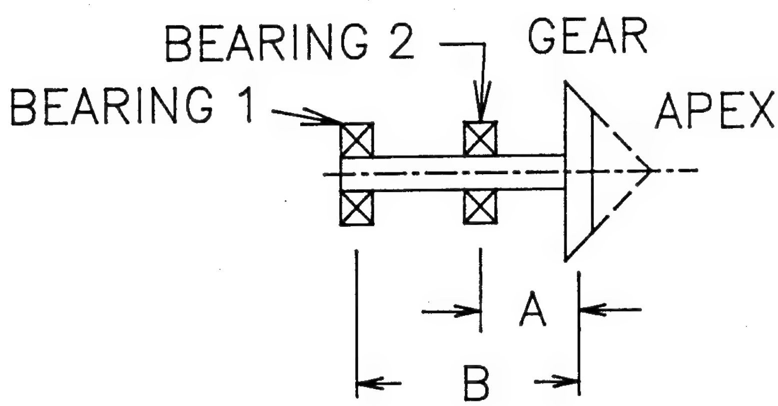
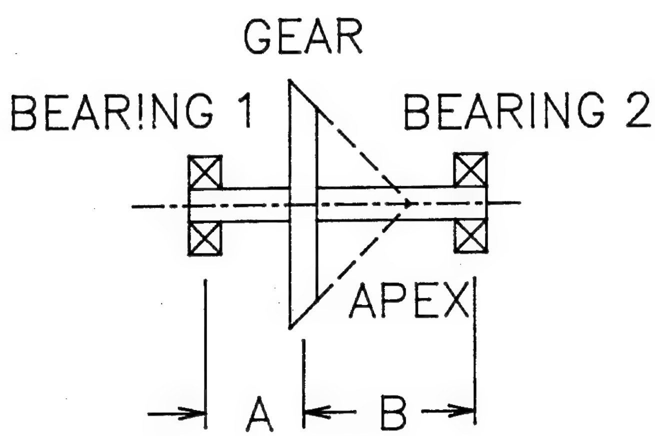


FIGURE 11 — SPIRAL BEVEL REDUCTION



OVERHUNG



STRADDLE

FIGURE 12 – SPIRAL BEVEL SHAFT SUPPORT MOUNTINGS

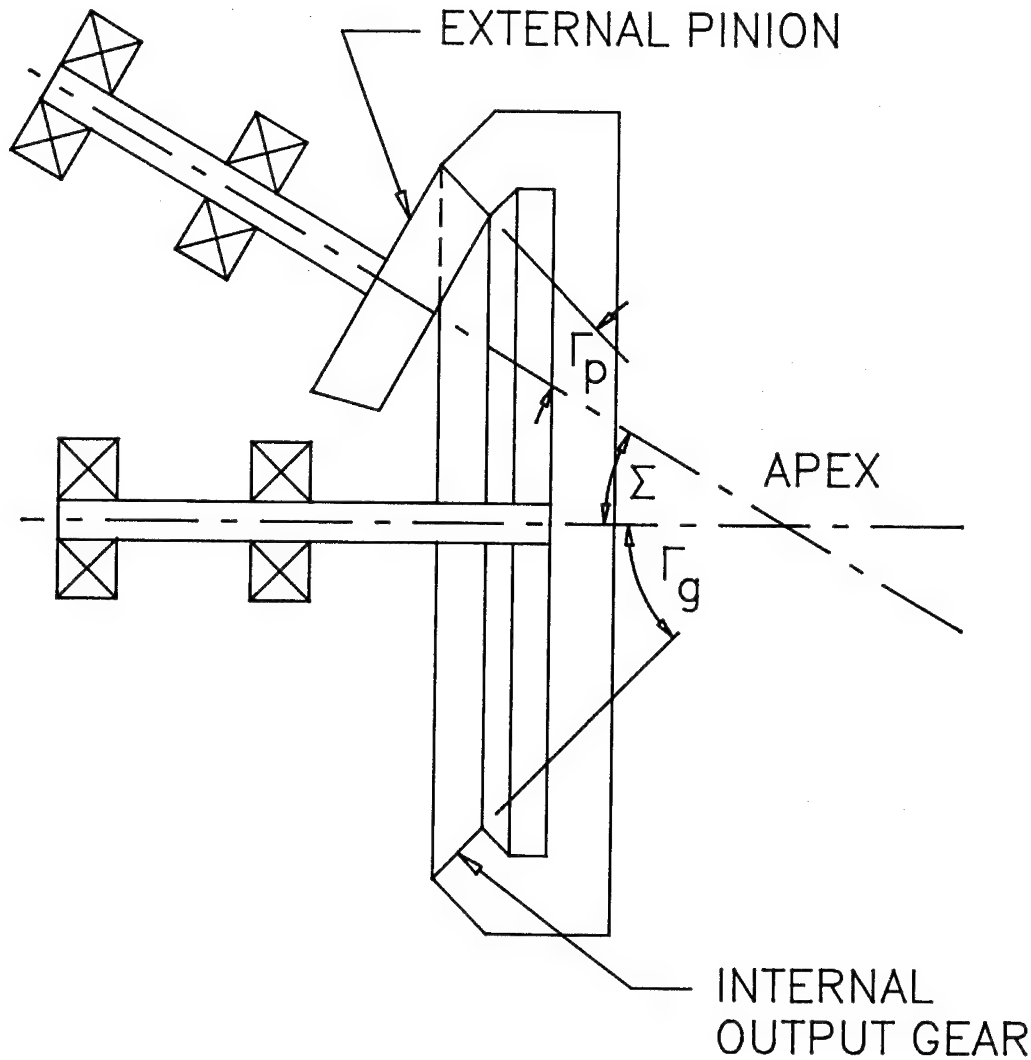


FIGURE 13 — SPIRAL BEVEL INTERNAL OUTPUT GEAR

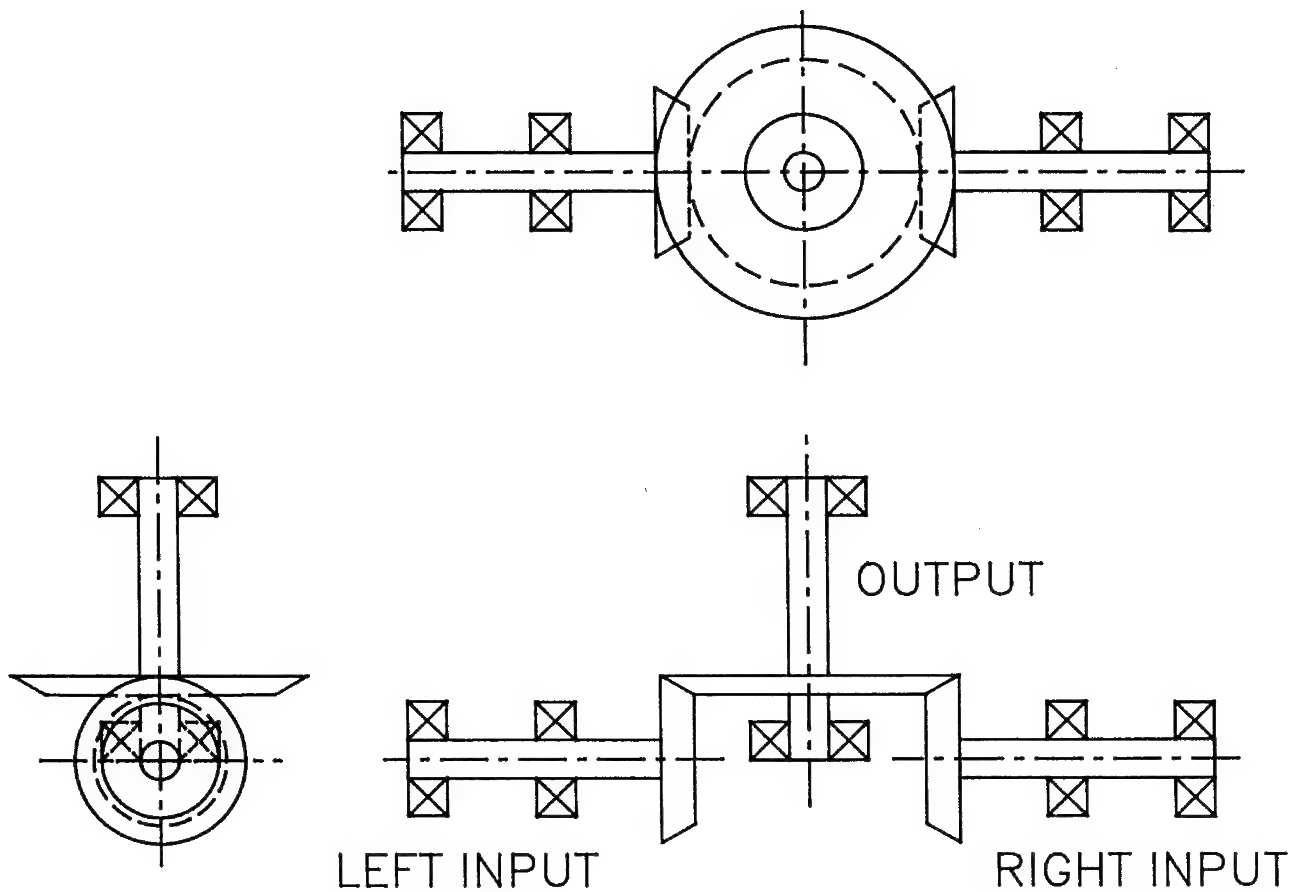


FIGURE 14 — DUAL SPIRAL BEVEL REDUCTION

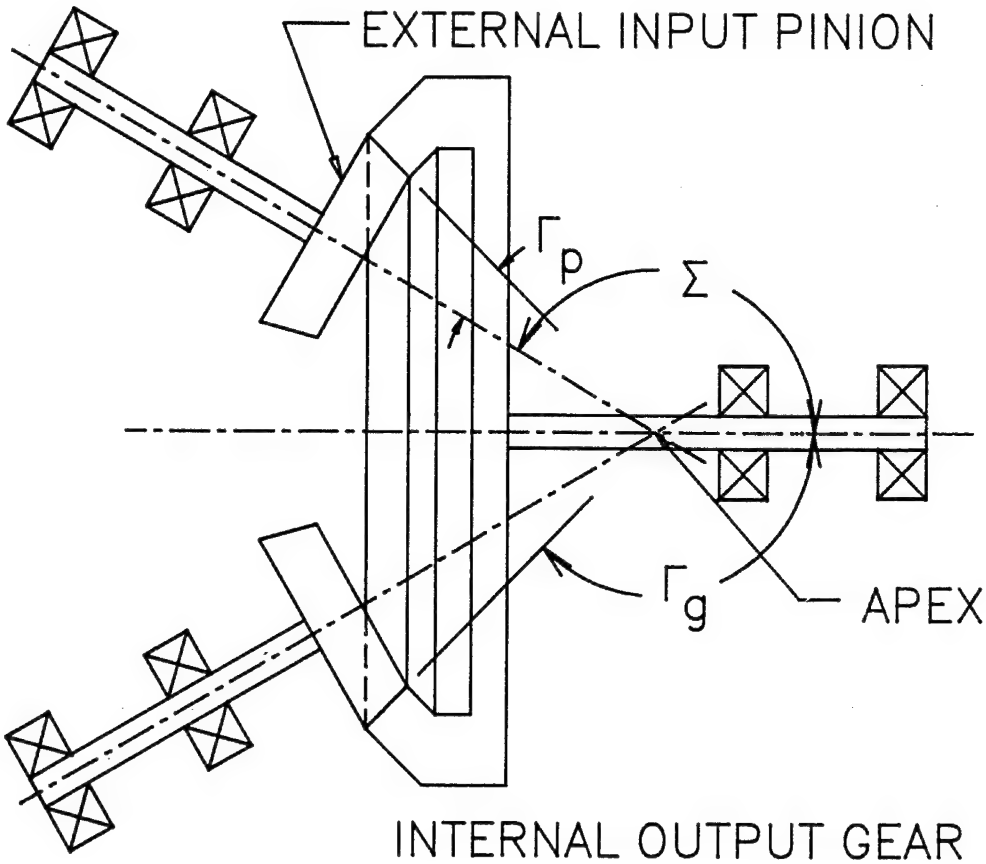


FIGURE 15 — DUAL SPIRAL BEVEL REDUCTION
WITH AN INTERNAL OUTPUT GEAR

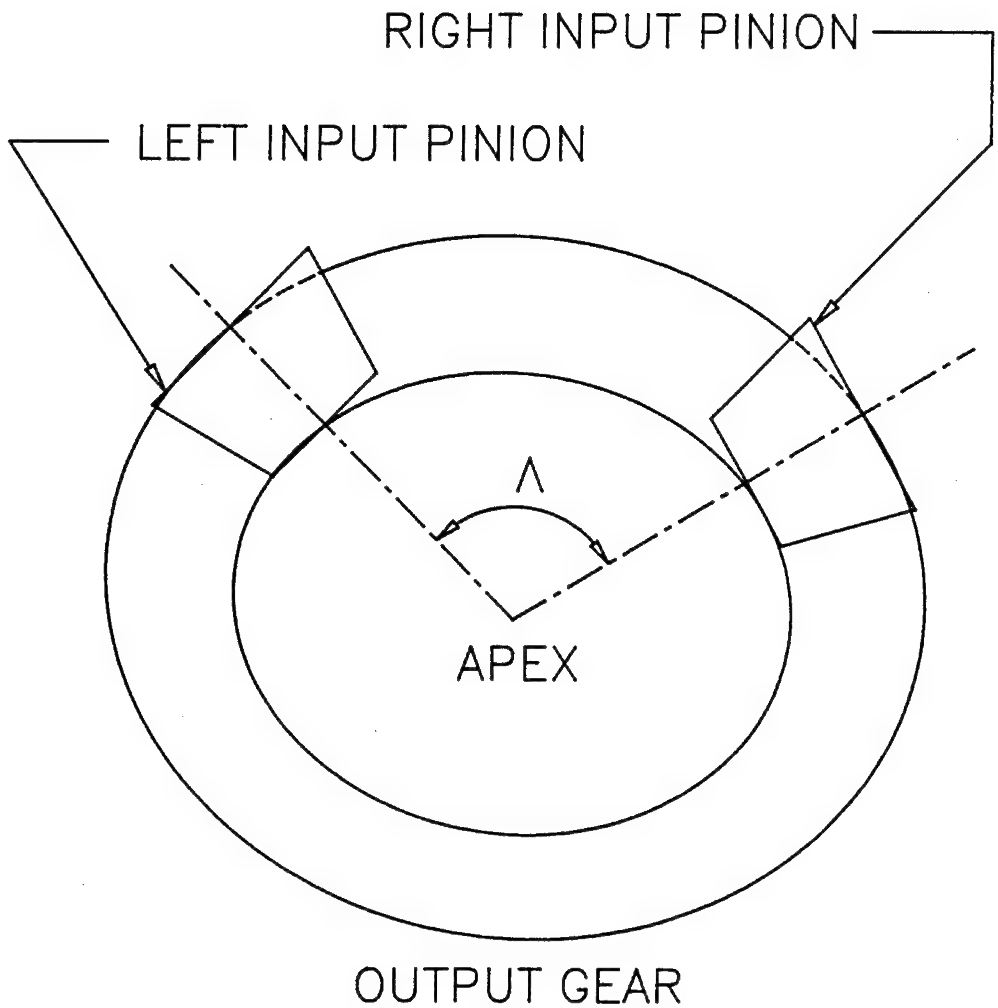


FIGURE 16 — DUAL SPIRAL BEVEL PINION SEPARATION ANGLE

respect to the output gear as defined by the angle Λ between the input pinions in the plane of their axes. The distances A and B along with the angle Λ again specify the geometry. Here again, the analysis assumes that one of the bearings takes the thrust load, or that both the bearings share the thrust load equally.

Transmission Systems

Although single unit transmissions may be analyzed, up to twenty-five transmission units may be combined in series for analysis or a two branch transmission may be analyzed with the dual spiral-bevel unit acting as a combining element with a twenty-five unit limit for one branch. The dual spiral bevel need not be the first unit in the transmission, but its dual inputs are assumed to be equal, so the two branches that feed it need to be identical.

In combining units which have two bearings on the output shaft of the driving unit and two bearings on the input shaft of the driven unit, four bearings are not placed on the same shaft. Instead, the bearings on the output shaft of the driving unit are ignored and the input shaft of the driven unit has the loads from both the output gear of the driving unit and the input gear of the driven unit superimposed on the two bearings of this input shaft. Where either unit transmission is a reverted spur reduction or a single-plane reduction, the other unit's interfacing shaft is analyzed as though it were the input or output shaft of a single unit transmission, since no bearings or gear loads are present.

The analysis assumes that there are no transmission losses between individual unit transmissions. The results for the overall transmission are written to the output file from the main program.

PROGRAM USE

To use the program on a PC, one must have the file "TLIFE.EXE" in the current directory or in a directory which is in the current path. An input data file must also be present in an active directory in the machine. One begins the program by typing TLIFE at the prompt and pressing the "enter" key. The program responds by listing the unit transmission options on the screen and requesting the prefix for the input data file. This prefix should include the full path to the data file if it is not in the present directory and may include up to 26 characters. The program then reads the input file, performs the analysis and writes an output file with the same prefix and the extension ".out" to the same directory which contains the input file. At the completion of this transmission analysis, the user is asked if another case is to be run. If the user responds with a "y", then he or she is prompted for the name of the input file as before. If the response is "n", the program is terminated and the user is returned to the operating system prompt. If any other response is given, the request to run another case is repeated.

Input Data File

An input data file describes the transmission to be analyzed. The data must be entered into an ASCII file, structured by line, but with a free format on each line. If multiple transmission units are to be analyzed as a system, then the units should be entered in order, with the highest speed unit first and the lowest speed unit last. If a dual spiral-bevel unit is included, only one input branch need be entered, as both input branches should be identical.

The first data line, labeled line 0 in group A in the program's ".DOC" file, should contain data which describes the overall transmission. Appendix A lists this ".DOC" file which is also included at the start of the program's source listing. The data in the first group is:

1) NUMT = (1,..,25), the integer number of unit transmissions in the system to be analyzed;
2) MET = (1/2), an integer indicating the analysis system of units - 1 - SI metric or 2 - English inch;
3) TORQUE = the system input torque in N - m or lb - in;
4) SPEED = the system input speed in RPM; and
5) NDIR = (1/2), an integer to indicate the direction of input torque and rotation looking into the transmission at the input shaft - 1 - counter-clockwise - 2 - clockwise.

Following this line, are similar sets of data lines - one set for each unit in the overall transmission.

Each set of data lines for the separate unit transmissions should start with a line 1 of group B containing just one integer - the number of the unit transmission as listed on page 7. Following this data line will be a series of lines which vary for each unit. The next line will be a transmission unit configuration line which is called group C. Four other different types of data lines will follow in varying numbers and location for each unit transmission corresponding to its complexity. The data group types are:

group B - unit configuration number,
group C - unit configuration properties,
group D - gear mesh properties,

group E - individual gear properties,

group F - shaft geometry and properties, and

group G - individual bearing properties.

Group C is the most configuration dependent group. It should contain up to four of the following seven items:

1) NRING = (1/2) - an integer switch to identify the presence of an internal (ring) gear as the output gear of the unit - 1 - external output gear - 2 - internal (ring) output gear;

2) NP = (1,..,7) - an integer for the number of parallel load paths, such as planet gears, in the unit;

3) NARM = (1/2) - an integer to indicate a units output member - 1 - last gear with the arm fixed or - 2 - arm with last gear fixed;

4) NOUT = (1/2) - an integer to locate the output shaft of a parallel compound reduction - 1 - on the same side of the intermediate shafts as the input shaft or - 2 - on the opposite side of the intermediate shafts from the input shaft;

5) SIGMA = the shaft angle in degrees. This angle has different meanings for the different unit transmissions which contain it. It is measured - counter-clockwise from the input to the output about the intermediate shaft as seen from the input for a compound reduction, between two intermediate shafts about the input shaft for the parallel compound reduction, or counter-clockwise from

the input shaft to the output shaft about the apex for the spiral-bevel reductions;

6) LAMBDA = the shaft angle between the two spiral pinions in degrees; and

7) BETA = the angle between the radial force on the output gear of the last unit transmission and the radial force on the input gear of this unit transmission measured counter-clockwise as seen from the high speed side in degrees.

For the spur and helical single-mesh cases (1 and 2), group C will contain one item - the integer NRING. For the spur and helical compound-reduction cases (3 and 4) and the spiral-bevel reduction (10), group C will contain two items - the integer NRING and the shaft angle SIGMA. For the spur and helical parallel compound reduction cases (5 and 6), group C will contain three items - the integers NRING and NOUT and the shaft angle SIGMA. For the spur and helical reverted cases (7 and 8) and the single-plane spur case (9), group C will contain three integer items - NRING, NP and NARM. And for the dual spiral-bevel case (11), group C will contain three items - NRING, SIGMA and LAMBDA. The seventh item, BETA, will be added for any unit transmission other than the reverted and single-plane reductions which follows a unit transmission other than a reverted or single-plane reduction in a series transmission. The angle BETA is needed to properly combine the shaft loads on the superimposed output and input shafts. The remaining data groups will be less configuration dependent.

Group D should contain the data for each gear mesh in the transmission unit. This group will contain three items for all gear meshes

and two more for the helical and spiral-bevel gear meshes. These items are:

1) the normal module in mm (SI) or the normal diametral pitch in in^{-1} (Eng) for spur or helical gears or the back cone distance, Ao, in mm (SI) or in (Eng) for spiral-bevel gears;
2) the normal pressure angle in degrees;
3) the axial face width in mm (SI) or in (Eng);
4) the helix or spiral angle in degrees; and
5) an integer to indicate the hand of the pinion helix or spiral angle - 1 - right hand, - 2 - left hand or - 3 - herringbone.

Group E should contain five items for each gear in the unit:

1) the number of teeth on the gear as an integer;
2) the gear addendum in mm (SI) or in (Eng), measured in the back plane for spiral bevel gears;
3) the gear Weibull slope;
4) the gear surface strength in MPa (SI) or ksi (Eng); and
5) the gear surface load-life exponent.

Group F should contain information which describes the support geometry of a shaft. This group will have four or five items depending on the type of shaft it describes. An input or output shaft will only require four while an intermediate shaft will need five items. The first two items are integer switches and the last two or three items are distances which are shown in Figure 5 for input and output shafts and Figure 7 for intermediate shafts. These items are:

1) NTHR = (0,..,3) an integer describing which bearing supports the axial shaft thrust load,

   0 - neither bearing supports thrust,

   1 - only bearing 1 supports thrust,

   2 - only bearing 2 supports thrust, or

   3 - both bearings share the thrust equally;

2) NCASE = (1,..,4) an integer describing the mounting,

   1 - straddle mounting of a single gear or both gears,

   2 - overhung mounting of a single gear or both gears,

   3 - straddle mounting of the first gear and overhung mounting of the second gear, and

   4 - overhung mounting of the first gear and straddle mounting of the second gear;

3) the distance A for single-gear support or C for two gear support in mm (SI) or in (Eng);

4) the distance B for single-gear support or D for two gear support in mm (SI) or in (Eng); and

5) the distance E for two gear support in mm (SI) or in (Eng).

Group G should contain the information which describes a bearing. The first five items are required for all bearings, the sixth and seventh are needed for ball bearings that support thrust, and the eighth and ninth are needed for tapered roller bearings. Note that the basic dynamic capacity for a double-row tapered roller bearing should be the radial capacity of a single row of the bearing. The program will double this

capacity to model the full bearing radial capacity if the load is purely radial. Otherwise, the axial load biases the radial support in the bearing to one side, so only the single-row capacity is required. The items for group G are:

1) ITY = (1,..,5) an integer identifying the bearing type -

   1 - single-row ball,

   2 - double-row ball,

   3 - single-row straight roller,

   4 - double-row straight roller, and

   5 - double-row tapered roller;

2) the basic dynamic capacity for one-million rotations in kN (SI) or lbs (Eng);

3) the Weibull slope;

4) the life adjustment factor;

5) the load adjustment factor for race rotation;

6) the static capacity of the ball bearing in kN (SI) or lbs (Eng);

7) the unmounted bearing contact angle in degrees;

8) the axial pre-load of the tapered bearing in kN (SI) or lbs (Eng); and

9) the radial to axial thrust capacity ratio.

These data groups are assembled in different orders and numbers to describe the different unit transmissions.

For cases 1, 2, 10 and 11 - the first two unit transmissions, the spur and helical single-mesh transmissions, and for the last two unit

transmissions, the spiral-bevel and dual spiral-bevel gear reductions, eleven data lines are required:

1. group B describing the unit number,
2. group C describing the transmission,
3. group D describing the gear mesh,
4. group E describing the input pinion,
5. group F describing the pinion support shaft,
6. group G describing the first pinion bearing,
7. group G describing the second pinion bearing,
8. group E describing the output gear,
9. group F describing the gear support shaft,
10. group G describing the first gear bearing, and
11. group G describing the second gear bearing.

For cases 3, 4, 5, 6 and 8 - the next four unit transmissions, the spur and helical compound reductions and the spur and helical parallel compound reductions, and for the reverted helical reduction, seventeen data lines are required:

1. group B describing the unit number,
2. group C describing the transmission,
3. group D describing the first gear mesh,
4. group D describing the second gear mesh,
5. group E describing the input pinion,
6. group F describing the pinion support shaft,
7. group G describing the first pinion bearing,
8. group G describing the second pinion bearing,
9. group E describing the first intermediate gear,

10. group E describing the second intermediate gear,
11. group F describing the intermediate gear support shaft,
12. group G describing the first intermediate shaft bearing,
13. group G describing the second intermediate shaft bearing,
14. group E describing the output gear,
15. group F describing the output gear support shaft,
16. group G describing the first output gear bearing, and
17. group G describing the second output gear bearing.

For case 7 - the reverted spur reductions, eleven data lines are required:

1. group B describing the unit number,
2. group C describing the transmission,
3. group D describing the input pinion mesh,
4. group D describing the output gear mesh,
5. group E describing the input pinion,
6. group E describing the intermediate gear meshing with the pinion,
7. group E describing the intermediate gear meshing with the output gear,
8. group F describing the intermediate gear support shaft,
9. group G describing the first bearing,
10. group G describing the second bearing, and
11. group E describing the output gear.

For case 9 - the single-plane reduction, nine data lines are required, which are:

1. group B describing the unit number,
2. group C describing the transmission,

3. group D describing the input pinion mesh,
4. group D describing the output ring gear mesh,
5. group E describing the input pinion sun gear,
6. group E describing the planet gear meshing with the sun,
7. group E describing the planet meshing with the output ring gear,
8. group G describing the planet bearing, and
9. group E describing the output ring gear.

When two unit transmissions, for which the output shaft of the first transmission and the input shaft of the second transmission both have two support bearings, are joined in series it is assumed that only two of the four bearings are present in the combined transmission and that these bearings support both the output gear of the first transmission and the input gear of the second transmission. The two bearing descriptions and the distance between them should be the same in the data groups for each transmission.

Output Data File

For each case run, an ASCII output data file is written to the directory which contains the input data file. This output data file has three basic parts: 1) a power, torque and speed transmission summary; 2) an analysis of the components including their geometry, loading and dynamic capacities; and 3) an output dynamic capacity, ninety-percent reliability life and mean life summary for the components and the transmission. For transmissions composed of several unit transmissions, an overall input power, torque and speed transmission summary and output transmission dynamic capacity and life summary are added.

APPLICATIONS

Consider three example transmissions: 1) a single-mesh spiral-bevel reduction, 2) a compound helical-gear reduction and 3) two single-mesh helical-gear reductions in series. The second and third cases are the same transmission, but their analyses by the program are different to illustrate the flexibility of the program and verify its consistency in analysis. For each example, the transmission will be described, its input data file will be constructed and its resulting output data file will be shown and explained.

Spiral Bevel

A schematic of the spiral-bevel gear example is shown in Figure 17. For this single-unit transmission example, the metric system of units is used. The transmission has an input torque of 250 N-m in the counter-clockwise direction and an input speed of 4,000 rpm. The transmission has an external output gear which is supported on a shaft which makes a 110 degree shaft angle with the input shaft. The gear mesh has a back cone distance of 120 mm, an axial face width of 30 mm, a 20 degree normal pressure angle and a 35 degree spiral angle with a right-hand spiral on the input pinion.

The 21 tooth input pinion meshes with a 42 tooth external output gear. The pinion has an addendum of 5.633 mm and the gear has an addendum of 2.749 mm. Both gears are made of high strength steel and have a surface material strength of 1500 MPa for a Hertzian-contact fatigue life of one million load cycles [13], a Weibull slope of 2.5 [5,6], and a load-life exponent of 8.93 [13]. The pinion and the output gear are mounted in overhung mountings with the first bearing closest to the gear and the second bearing furthest behind the gear. The bearings are 300 series with a 55 mm bore. On the input and output shafts, the first bearings are single-row straight-roller

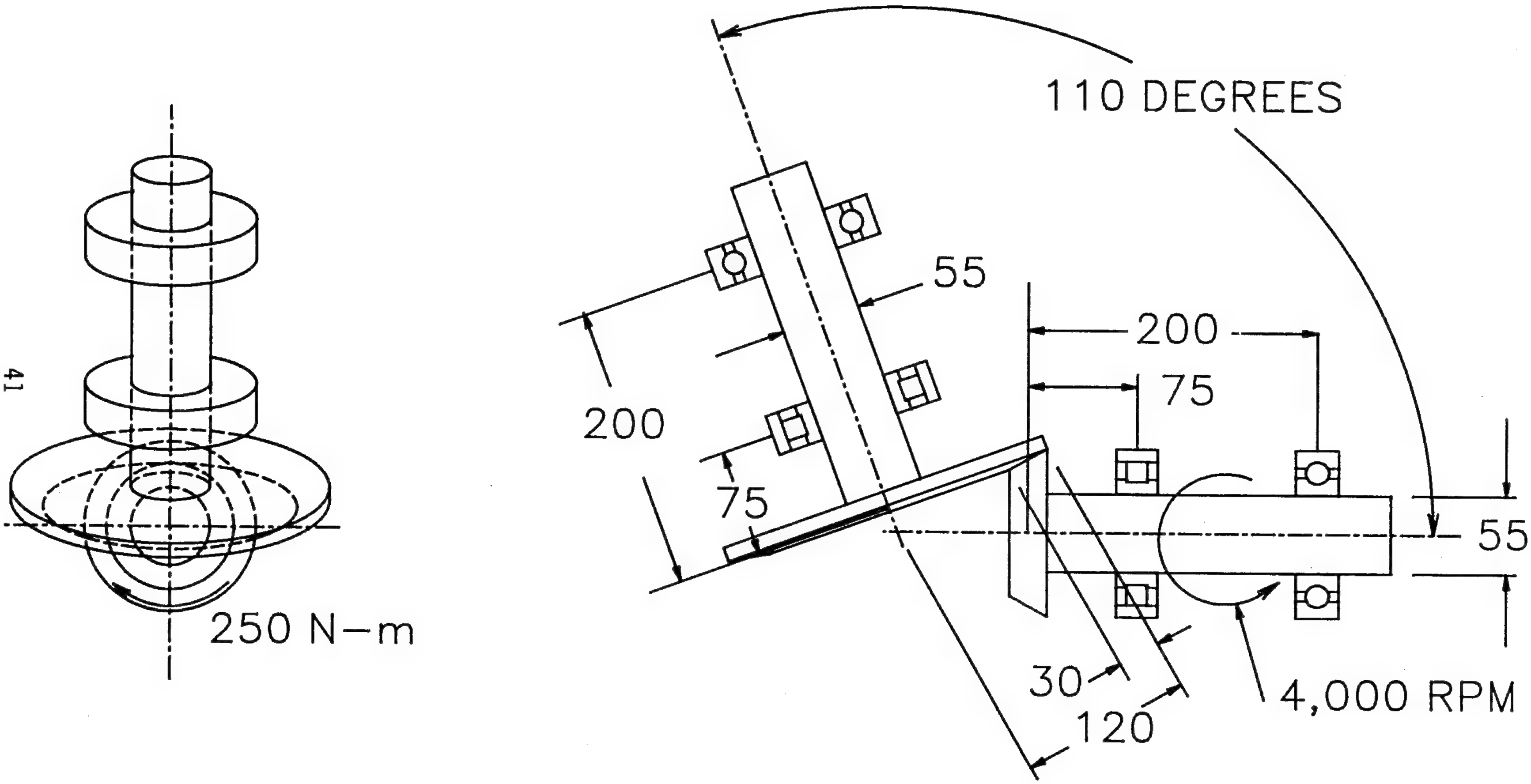


FIGURE 17 – SPIRAL BEVEL EXAMPLE

bearings and are located 75 mm from their respective gears. The second bearings are single-row ball bearings, which support the thrust loads and are 200 mm behind their respective gears. The bearing location distances are measured from the center of the gear face width to the bearing center. The pinion roller bearing has a dynamic capacity of 19.7 kN, and the pinion ball bearing has a dynamic capacity of 10.9 kN, a static capacity of 35 kN, and a 25 degree contact angle [14]. The dynamic capacity of the output gear roller bearing is 23.3 kN, while the output gear ball bearing has a dynamic capacity of 12.9 kN, a static capacity of 35 kN, and a 25 degree contact angle. The gear support bearings have higher dynamic capacities than their corresponding pinion support bearings due to their lower speed. The two roller bearings have a Weibull slope of 9/8 while the two ball bearings have a Weibull slope of 10/9 [15,16]. All four bearings have a life adjustment factor of 6.0 and a load adjustment factor of 1.0 for race rotation.

Directions for writing the input file can be found in the section, "PROGRAM USE," which precedes these examples and in the ASCII file "TLIFE.DOC" which accompanies the program and is listed in Appendix A. The ".DOC" file information is given at the start of the source code file "TLIFE.FOR" as well.

For the spiral-bevel reduction, twelve data lines are required - lines 0 through 11 as described in the ".DOC" file and listed on page 36 of this report. Table 1 is a summary of this input data, listed in the order in which it is needed for the input data file. Table 2 is the input data file which describes this transmission to the program. The twelve sections in Table 1 represent the twelve lines of data in the input file of Table 2, whereby each entry of the table corresponds to a single value in the input file for the TLIFE program. The first section lists the overall transmission system data:

TABLE 1

SPIRAL BEVEL GEAR UNIT INPUT DATA

Overall Transmission	
Number of unit transmissions	1
Metric system of units	1
System input torque	250.0 N-m
System input speed	4000.0 RPM
Counter-clockwise input torque and rotation	1
Unit Number	
Spiral-bevel gear unit number	10
Unit Configuration Properties	
External output gear	1
Shaft angle	110.0 degrees
Gear Mesh Properties	
Back cone distance	120.0 mm
Normal pressure angle	20.0 degrees
Axial face width	30.0 mm
Spiral angle	35.0 degrees
Right-hand pinion spiral	1
Pinion Properties	
Number of teeth on the pinion	21
Back cone addendum	5.633 mm
Gear Weibull slope	2.5
Gear surface material strength	1500.0 MPa
Gear surface load-life exponent	8.93
Pinion Support Shaft	
Only bearing 2 supports the thrust load	2
Overhung mounting	2
Distance A	75.0 mm
Distance B	200.0 mm
Pinion Bearing 1	
Single-row straight-roller bearing	3
Basic dynamic capacity	19.7 kN
Weibull slope	1.125
Life adjustment factor	6.0
Load adjustment factor for race rotation	1.0
Pinion Bearing 2	
Single-row ball bearing	1
Basic dynamic capacity	10.9 kN
Weibull slope	1.111
Life adjustment factor	6.0
Load adjustment factor for race rotation	1.0
Static capacity of ball bearing	35.0 kN
Bearing contact angle	25.0 degrees

TABLE 1 CONTINUED

Output Gear Properties	
Number of teeth on the output gear	42
Back cone addendum	2.749 mm
Gear Weibull slope	2.5
Gear surface material strength	1500.0 MPa
Gear surface load-life exponent	8.93
Gear Support Shaft	
Only bearing 2 supports the thrust load	2
Overhung mounting	2
Distance A	75.0 mm
Distance B	200.0 mm
Gear Bearing 1	
Single-row straight-roller bearing	3
Basic dynamic capacity	23.3 kN
Weibull slope	1.125
Life adjustment factor	6.0
Load adjustment factor for race rotation	1.0
Gear Bearing 2	
Single-row ball bearing	1
Basic dynamic capacity	12.9 kN
Weibull slope	1.111
Life adjustment factor	6.0
Load adjustment factor for race rotation	1.0
Static capacity of ball bearing	35.0 kN
Bearing contact angle	25.0 degrees

TABLE 2

SPIRAL BEVEL GEAR UNIT INPUT DATA FILE

1	1	250.0	4000.0	1		
10						
1	110.0					
120.	20.	30.0	35.	1		
21	5.633	2.5	1500.0	8.93		
2	2	75.	200.			
3	19.7	1.125	6.	1.0		
1	10.9	1.1111	6.	1.0	35.0	25
42	2.749	2.5	1500.0	8.93		
2	2	75.	200.			
3	23.3	1.125	6.	1.0		
1	12.9	1.1111	6.	1.0	35.0	25

the number of unit transmissions, the system of units, and the input torque, speed and direction. The second section identifies the unit transmission type. The third section gives the overall unit properties specific to the spiral-bevel reduction which are: the type of output gear and the shaft angle between the input pinion and the output gear. The fourth section lists the gear mesh properties: the back cone distance, the normal pressure angle, the axial face width, and the spiral angle and its direction on the pinion.

The remaining sections describe the individual gears, shaft configurations and bearings. Section five gives the pinion properties: number of teeth, addendum, Weibull slope, surface material strength for one million load cycles and load-life exponent. Section six gives the pinion support shaft geometry: the thrust bearing location, the type of mounting, and the distances to the two bearings. Section seven gives the data for the first bearing which is a straight roller: the type of bearing, its basic dynamic capacity, its Weibull slope, and its life and load adjustment factors. Section eight gives the data for the second bearing which is a ball bearing: the type of bearing, its basic dynamic capacity, its Weibull slope, its life and load adjustment factors, its static capacity and its contact angle. Section nine gives the gear properties, ten gives the gear support geometry data and eleven and twelve give the data which describe the gear support bearings.

For TLIFE to perform properly, each input file value must be entered in the order depicted in Table 2. In performing its analysis, the program assumes that the roller bearings have load-life exponents of 10/3 and the ball bearings have load-life exponents of 3.0 [15,16].

The program is run by typing TLIFE when the machine prompt is in the directory which contains the file "TLIFE.EXE". The program will ask for the name of the input file prefix. If the input file is in the same directory, the eight or fewer letter prefix will be sufficient to describe the data file to the program. If the input file is in another directory, one should type the full path to the file as part of the prefix. The program will execute and write an output file with the same prefix and the extension ".OUT" in the same directory in which the input file is located. The program then prompts the user with the question "Do you wish to run another case?" If the answer is Y or y for yes, the program will then request the name of another input file. If the answer is N or n for no, the program will stop and return control to the command processor.

The output data file for this example is listed in Appendix B. This file repeats the input information, includes geometry and load analysis data and tabulates the capacities and lives of the transmission components and system. The first block of information lists the input and output speeds of 4,000 and 2,000 RPM, the speed reduction ratio of 2, the transmitted power of 104.72 kW, the input torque and direction of 250 N-m - counter-clockwise, and the output torque and direction of 500 N-m - clockwise.

In the next block of information, the file describes: the pinion support geometry; the two bearings' basic capacities, loads and adjusted dynamic capacities in bearing load units; and the pinion gear geometry, contact ratios, loads and adjusted dynamic capacity in gear load units. Following this is a similar block of data for the output gear. Figure 18 depicts the loads on the input pinion and support bearings for this example as tabulated in the output file. Negative bearing forces have the same direction as the

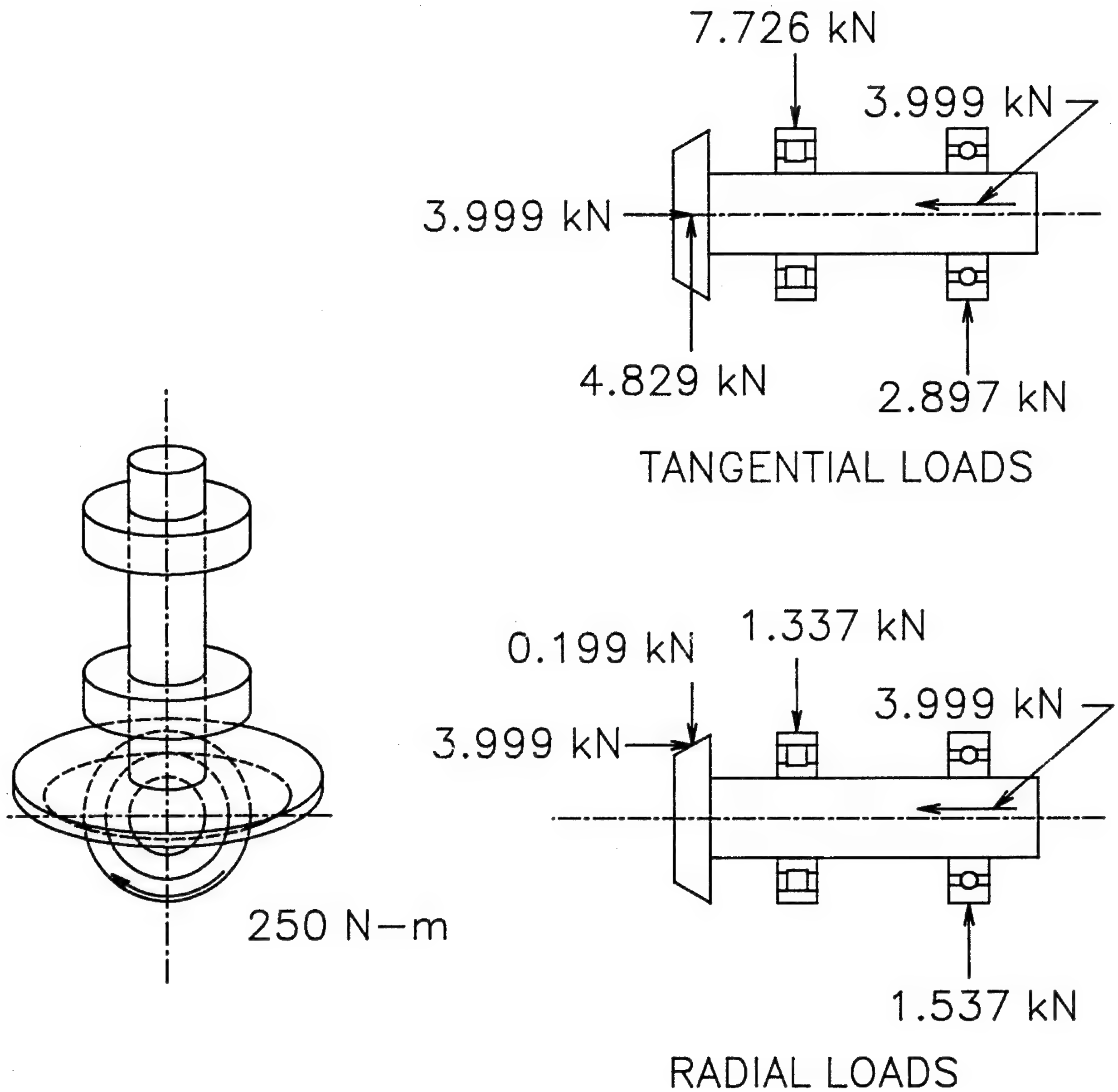


FIGURE 18 – INPUT PINION SHAFT LOADING
FOR SPIRAL BEVEL EXAMPLE

supported gear force. Figure 19 shows similar information for the output gear and its support bearings. Table 3 lists these loads and Table 4 lists the output dynamic capacities, L_{10} lives and mean lives of the components and the transmission as summaries of the information to be found in the output file.

In this example one can see the strong influence of the input ball bearing capacity on the transmission service life. One can also see that the service life is greater at 1,146 hours for replacing the entire transmission than 1,022 hours for replacing only the failed component in a maintenance session. By replacing the entire transmission, one replaces components with partial damage in addition to the failed component - thus extending the service life of the repaired transmission at a larger material cost per maintenance. These two lives are close to each other, due to the short life of the input shaft ball bearing relative to the lives of the other components in the transmission.

<u>Compound Helical</u>

A compound helical-gear unit transmission is shown in Figure 20. For this single-unit transmission, the English inch system of units is used. The transmission has an input torque of 600 lb-in in the counter-clockwise direction, and an input speed of 5,000 rpm. The transmission has an external output gear and a 30 degree shaft angle from the location of the input shaft to that of the output shaft. The first gear mesh has a diametral pitch of 6, a 0.75 inch face width, a 30 degree helix angle, and a right-hand helix on the pinion. The second gear mesh has a diametral pitch of 4, a 20 degree normal pressure angle, an axial face width of 1.25 inches, and a 25 degree helix angle with a right-hand helix on the pinion.

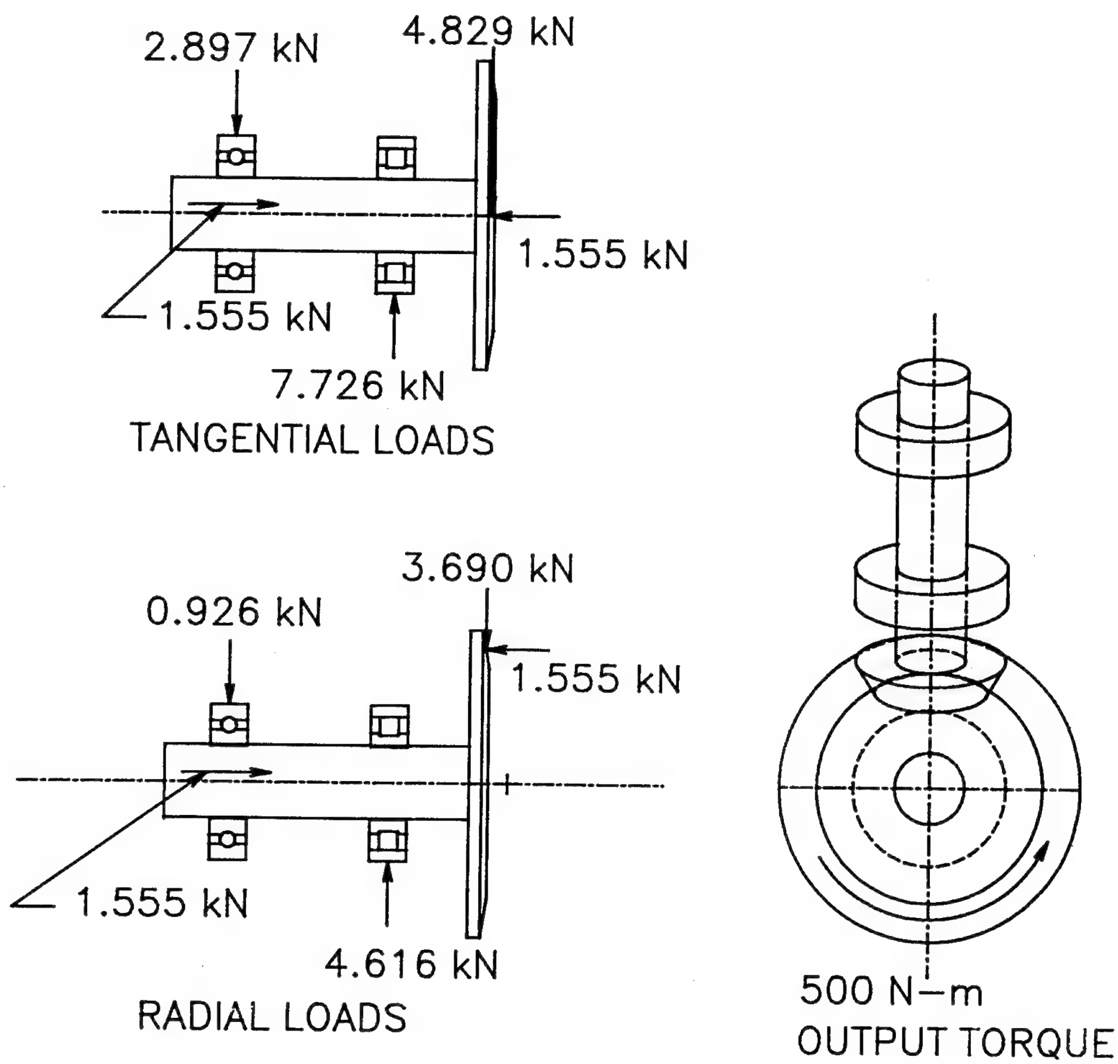


FIGURE 19 – OUTPUT GEAR SHAFT LOADING FOR SPIRAL BEVEL EXAMPLE

TABLE 3

SPIRAL BEVEL UNIT COMPONENT FORCES

COMPONENT	AXIAL FORCE (kN)	RADIAL FORCE (kN)	TANGENTIAL FORCE (kN)
Input Pinion	3.999	0.199	4.829
Input Roller Bearing	0.	-1.337	7.726
Input Ball Bearing	3.999	1.537	-2.897
Output Gear	1.555	3.690	4.829
Output Roller Bearing	0.	4.616	7.726
Output Ball Bearing	1.555	-0.926	-2.897

TABLE 4

SPIRAL BEVEL UNIT CAPACITIES AND LIVES

COMPONENT	DYNAMIC CAPACITY (N-m)	L10 LIFE IN MILLION OUTPUT ROTATIONS	MEAN LIFE (hours)
Input Pinion	2,772.36	4,393,803.	82,089,460.
Input Roller Bearing	1,752.48	62.73	3,429.32
Input Ball Bearing	1,629.36	34.60	1,939.40
Output Gear	2,904.53	6,659,760.	124,424,400.
Output Roller Bearing	2,227.92	138.51	7,572.15
Output Ball Bearing	3,853.48	457.77	25,655.27
Transmission	1,313.67	20.66	1,146.13
Components			1,022.19

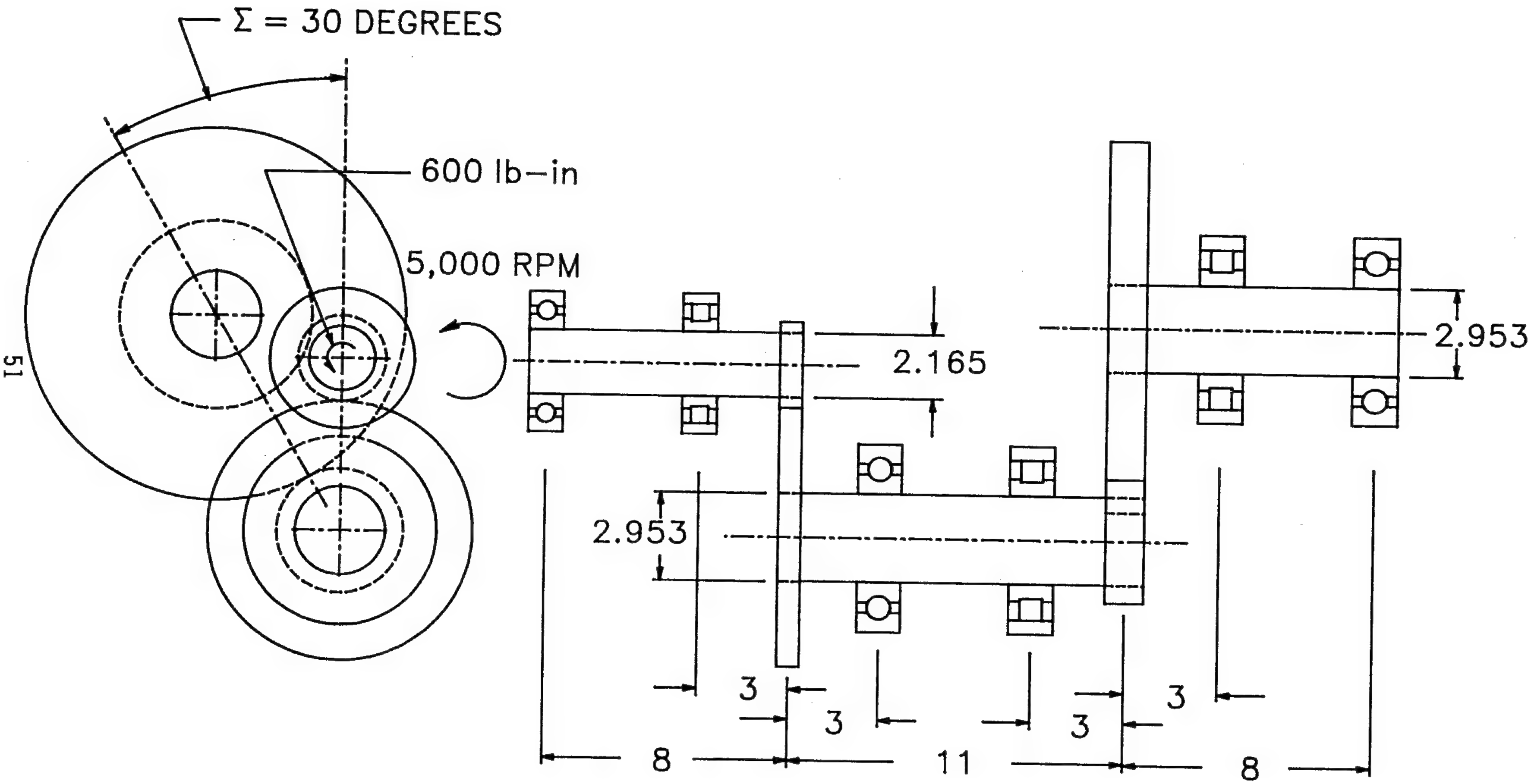


FIGURE 20 — COMPOUND HELICAL EXAMPLE

A 15 tooth input pinion meshes with a 45 tooth gear which drives a second 15 tooth pinion in mesh with an external output gear with 45 teeth. The first two gears have addenda of 0.167 inches while the second two gears have addenda of 0.25 inches. All the gears are made of high strength steel with a surface material strength of 220 ksi [13]. The Weibull slope for all four gears is 2.5 [6], and the load-life exponent of the gears is 8.93 [13].

The pinion is mounted in an overhung mounting with the second bearing furthest from the gear, which is the ball bearing, taking the thrust load. The single-row straight-roller bearing is 3 inches from the pinion and the single-row ball bearing is 8 inches from the pinion. These distances are measured from the pinion center to the respective bearing center. The two pinion bearings are 300 series with a 2.165 inch (55 mm) bore. The roller bearing has a dynamic capacity of 4,070 lbs, and the ball bearing has a dynamic capacity of 2,248 lbs, a static capacity of 7,880 lbs, and a 25 degree contact angle.

The intermediate gear is mounted in a double overhung mounting with the first bearing, the ball bearing, taking the thrust load. The single-row ball bearing is closest to the first intermediate gear, at a distance of 3 inches. The single-row straight-roller bearing is 3 inches from the second intermediate gear. The bearing locations are measured between the centers of the respective intermediate gears and bearings. A distance of 11 inches separates the centers of the two intermediate gears. The intermediate gear bearings are 300 series with a 2.953 inch (75 mm) bore. The intermediate ball bearing has a dynamic capacity of 4,160 lbs, a static capacity of 13,100 lbs, and a 25 degree contact angle. The intermediate roller bearing has a dynamic capacity of 8,810 lbs.

The output gear is mounted in an overhung mounting with the second bearing, the ball bearing, taking the thrust load. The single-row straight-roller bearing is 3 inches from the output gear and the single-row ball bearing is 8 inches away. Like the intermediate bearings, the output gear bearings are 300 series with a 2.953 inch (75 mm) bore. The output roller bearing has a dynamic capacity of 12,070 lbs, and the output ball bearing has a dynamic capacity of 5,690 lbs, a static capacity of 13,100 lbs, and a 25 degree contact angle. The pinion, intermediate, and output gear roller bearings all have a Weibull slope of 9/8. All the ball bearings have a Weibull slope of 10/9 and all six bearings have a life adjustment factor of 6 and a load adjustment factor for race rotation of 1.

Table 5 summarizes the input data for the compound helical-gear transmission, with the eighteen sections in Table 5 representing the eighteen lines of data in the input file, Table 6. Each entry of Table 5 corresponds to a single input value in this input file. As with the spiral-bevel case, the ".DOC" file and the section "PROGRAM USE" of this report describe the required data and its order for the input file.

The first section lists the overall transmission system data: the number of unit transmissions, the system of units, and the input torque, speed and direction. The second section gives the unit transmission type. The third section gives the overall unit properties specific to the compound helical reduction which are: the type of output gear and the shaft angle between the input and output shaft locations. The fourth section lists the gear mesh properties for the high-speed gear mesh: the normal diametral pitch, the normal pressure angle, the axial face width, and the helix angle and its hand

TABLE 5

COMPOUND HELICAL GEAR UNIT INPUT DATA

Overall Transmission	
Number of unit transmissions	1
English system of units	2
System input torque	600.0 lb-in
System input speed	5000.0 RPM
Counter-clockwise input torque and rotation	1
Unit Number	
Compound helical gear unit number	4
Unit Configuration Properties	
External output gear	1
Shaft angle	30.0 degrees
Gear Mesh 1 Properties	
Normal diametral pitch	6.0 1/inch
Normal pressure angle	20.0 degrees
Axial face width	0.75 inches
Helix angle	30.0 degrees
Right-hand pinion helix	1
Gear Mesh 2 Properties	
Normal diametral pitch	4.0 1/inch
Normal pressure angle	20.0 degrees
Axial face width	1.25 inches
Helix angle	25.0 degrees
Right-hand pinion helix	1
Pinion Properties	
Number of teeth on the pinion	15
Addendum	0.167 in
Gear Weibull slope	2.5
Gear surface material strength	220.0 ksi
Gear surface load-life exponent	8.93
Pinion Support Shaft	
Only bearing 2 supports the thrust load	2
Overhung mounting	2
Distance A	3.0 inches
Distance B	8.0 inches
Pinion Bearing 1	
Single-row straight-roller bearing	3
Basic dynamic capacity	4070.0 lbs
Weibull slope	1.125
Life adjustment factor	6.0
Load adjustment factor for race rotation	1.0

TABLE 5 CONTINUED

Pinion Bearing 2

Single-row ball bearing	1
Basic dynamic capacity	2248.0 lbs
Weibull slope	1.1111
Life adjustment factor	6.0
Load adjustment factor for race rotation	1.0
Static capacity of ball bearing	7880.0 lbs
Bearing contact angle	25.0 degrees

First Intermediate Gear Properties

Number of teeth on the first intermediate gear	45
Addendum	0.167 in
Gear Weibull slope	2.5
Gear surface material strength	220.0 ksi
Gear surface load-life exponent	8.93

Second Intermediate Gear Properties

Number of teeth on the second intermediate gear	15
Addendum	0.25 in
Gear Weibull slope	2.5
Gear surface material strength	220.0 ksi
Gear surface load-life exponent	8.93

Intermediate Gear Support Shaft

Only bearing 1 supports the thrust load	1
Double overhung mounting	2
Distance C	3.0 inches
Distance D	11.0 inches
Distance E	3.0 inches

Intermediate Bearing 1

Single-row ball bearing	1
Basic dynamic capacity	4160.0 lbs
Weibull slope	1.1111
Life adjustment factor	6.0
Load adjustment factor for race rotation	1.0
Static capacity of ball bearing	13100.0 lbs
Bearing contact angle	25.0 degrees

Intermediate Bearing 2

Single-row straight-roller bearing	3
Basic dynamic capacity	8810.0 lbs
Weibull slope	1.125
Life adjustment factor	6.0
Load adjustment factor for race rotation	1.0

TABLE 5 CONTINUED

Output Gear Properties

Number of teeth on the output gear 45
Addendum .... 0.25 in
Gear Weibull slope 2.5
Gear surface material strength .... 220.0 ksi
Gear surface load-life exponent 8.93

Gear Support Shaft

Only bearing 2 supports the thrust load 2
Overhung mounting 2
Distance A .... 3.0 inches
Distance B .... 8.0 inches

Gear Bearing 1

Single-row straight-roller bearing 3
Basic dynamic capacity .... 12070.0 lbs
Weibull slope 1.125
Life adjustment factor 6.0
Load adjustment factor for race rotation 1.0

Gear Bearing 2

Single-row ball bearing 1
Basic dynamic capacity .... 5690.0 lbs
Weibull slope 1.1111
Life adjustment factor 6.0
Load adjustment factor for race rotation 1.0
Static capacity of ball bearing .... 13100.0 lbs
Bearing contact angle .... 25.0 degrees

TABLE 6

COMPOUND HELICAL GEAR UNIT INPUT DATA FILE

1	2	600.0	5000.0	1		
4						
1	30.0					
6.	20.	0.75	30.	1		
4.	20.	1.25	25.	1		
15	0.167	2.5	220.0	8.93		
2	2	3.	8.			
3	4070.	1.125	6.	1.0		
1	2248.	1.1111	6.	1.0	7880.0	25
45	0.167	2.5	220.0	8.93		
15	0.25	2.5	220.0	8.93		
1	2	3.	11.	3.		
1	4160.	1.1111	6.	1.0	13100.0	25
3	8810.	1.125	6.	1.0		
45	0.25	2.5	220.0	8.93		
2	2	3.	8.			
3	12070.	1.125	6.	1.0		
1	5690.	1.1111	6.	1.0	13100.0	25

on the input pinion. The fifth section gives the corresponding properties for the low-speed gear mesh.

The remaining sections describe the individual gears, shaft configurations and bearings. Section six gives the high-speed pinion properties: the number of teeth, the addendum, the Weibull slope, the material surface strength for one million load cycles and the load-life exponent. Section seven gives the input shaft properties: the thrust bearing location, the type of mounting, and the distances to the two bearings. Section eight gives the data for the first bearing which is a straight roller: the type of bearing, its basic dynamic capacity, its Weibull slope, and its life and load adjustment factors. Section nine gives the data for the second bearing which is a ball bearing: the type of bearing, its basic dynamic capacity, its Weibull slope, its life and load adjustment factors, its static capacity and its contact angle.

Section ten gives the properties for the high-speed output gear, which is also the first intermediate gear, in the same format as section six describes the input gear. Section eleven describes the low-speed input gear, which is also the second intermediate gear, with this same format. Section twelve describes the intermediate shaft support geometry: the thrust bearing location, the type of mounting, and three distances - from the first gear to the first bearing, between the two gears and from the second gear to the second bearing.

Section thirteen describes the first intermediate bearing, which is a ball bearing, and section fourteen gives the data for the second intermediate bearing, which is a straight roller bearing. Section fifteen gives the output gear properties, sixteen gives the output gear support geometry data and

seventeen and eighteen give the data which describe the output gear support bearings.

Once the input file has been created, TLIFE can be run by typing TLIFE as described on pages 45 and 46 of this report. Appendix C is a listing of the output file for this compound-mesh helical reduction example.

The compound helical-gear transmission unit transmits 47.6 HP from the input shaft, which turns counter-clockwise at 5,000 RPM with an input torque of 600 lb-in., to an output shaft which turns counter clockwise at 555 RPM with an output torque of 5,400 lb-in. The radial, axial, and tangential forces on the gears and bearings are summarized in Table 7. Negative bearing forces act in the same direction as the gear loads they support. Figures 21, 22, and 23 show the loads on the gears and bearings for the input, intermediate, and output shafts, which are summarized in Table 7 and reported in the output file of Appendix C.

Table 8 summarizes the primary output for the analysis of the compound helical-gear reduction. The table lists the dynamic capacity in lb-in, the L10 life in million output rotations, and the mean life in hours for the transmission and each reduction component. The low values of the bearings' lives compared to the lives of the pinion, intermediate gears, and output gear are due to the moderate loading on this transmission and the capacities of the different components. The service life of the transmission is affected greatly by the selected bearings. The example compound helical transmission has a mean time between repairs of 11,426 hours for complete transmission replacement at repair, and a mean time between overhauls of 9,831 hours for repair by failed component replacement.

TABLE 7

COMPOUND HELICAL GEAR UNIT COMPONENT FORCES

COMPONENT	AXIAL FORCE (lbs)	RADIAL FORCE (lbs)	TANGENTIAL FORCE (lbs)
Input Pinion	240.0	174.7	415.7
Input Roller Bearing	0.	210.2	665.1
Input Ball Bearing	240.0	-35.5	-249.4
Intermediate Gear #1	-240.0	174.7	415.7
Intermediate Gear #2	-405.7	349.4	870.1
Intermediate Ball Bearing	-645.7	17.1	1,062.2
Intermediate Roller Bearing	0.	691.4	1,659.6
Output Gear	-405.7	349.4	870.1
Output Roller Bearing	0.	1,062.7	1,392.1
Output Ball Bearing	-405.7	-713.3	-522.0

TABLE 8

COMPOUND HELICAL GEAR UNIT CAPACITIES AND LIVES

COMPONENT	DYNAMIC CAPACITY (lb-in)	L10 LIFE IN MILLION OUTPUT ROTATIONS	MEAN LIFE (hours)
Input Pinion	34,089	13,995,620	941,329,300
Input Roller Bearing	27,865	225	44,242
Input Ball Bearing	33,979	249	50,267
Intermediate Gear #1	36,701	27,056,090	1,819,761,000
Intermediate Gear #2	39,218	48,929,250	3,290,924,000
Intermediate Ball Bearing	26,642	120	24,231
Intermediate Roller Bearing	32,646	379	74,610
Output Gear	42,223	94,589,140	6,361,955,000
Output Roller Bearing	64,052	3,505	689,796
Output Ball Bearing	63,167	1,601	322,951
Transmission	19,753	57	11,426
Components			9,831

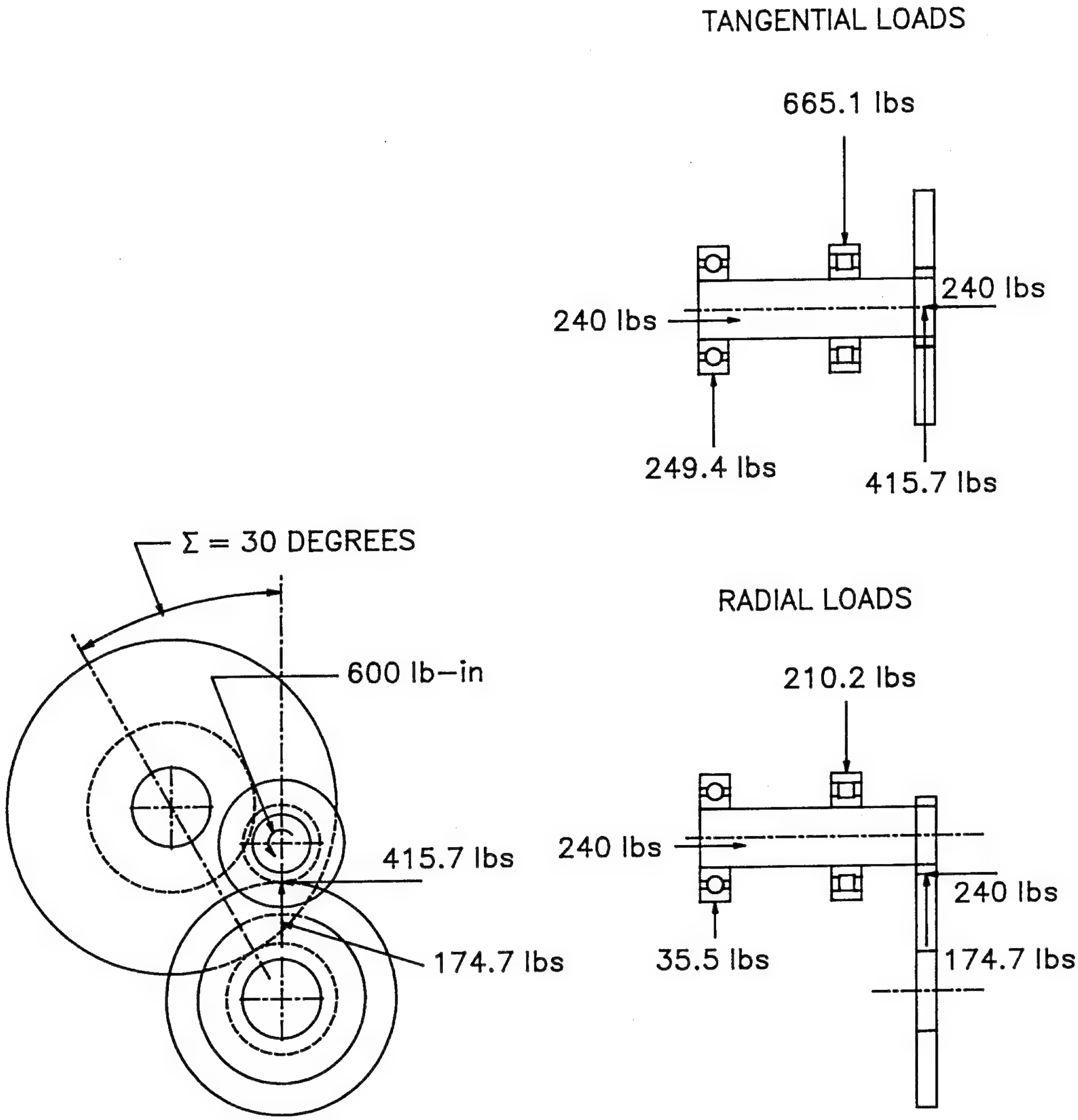


FIGURE 21 – INPUT PINION SHAFT LOADING FOR COMBINED HELICAL EXAMPLE

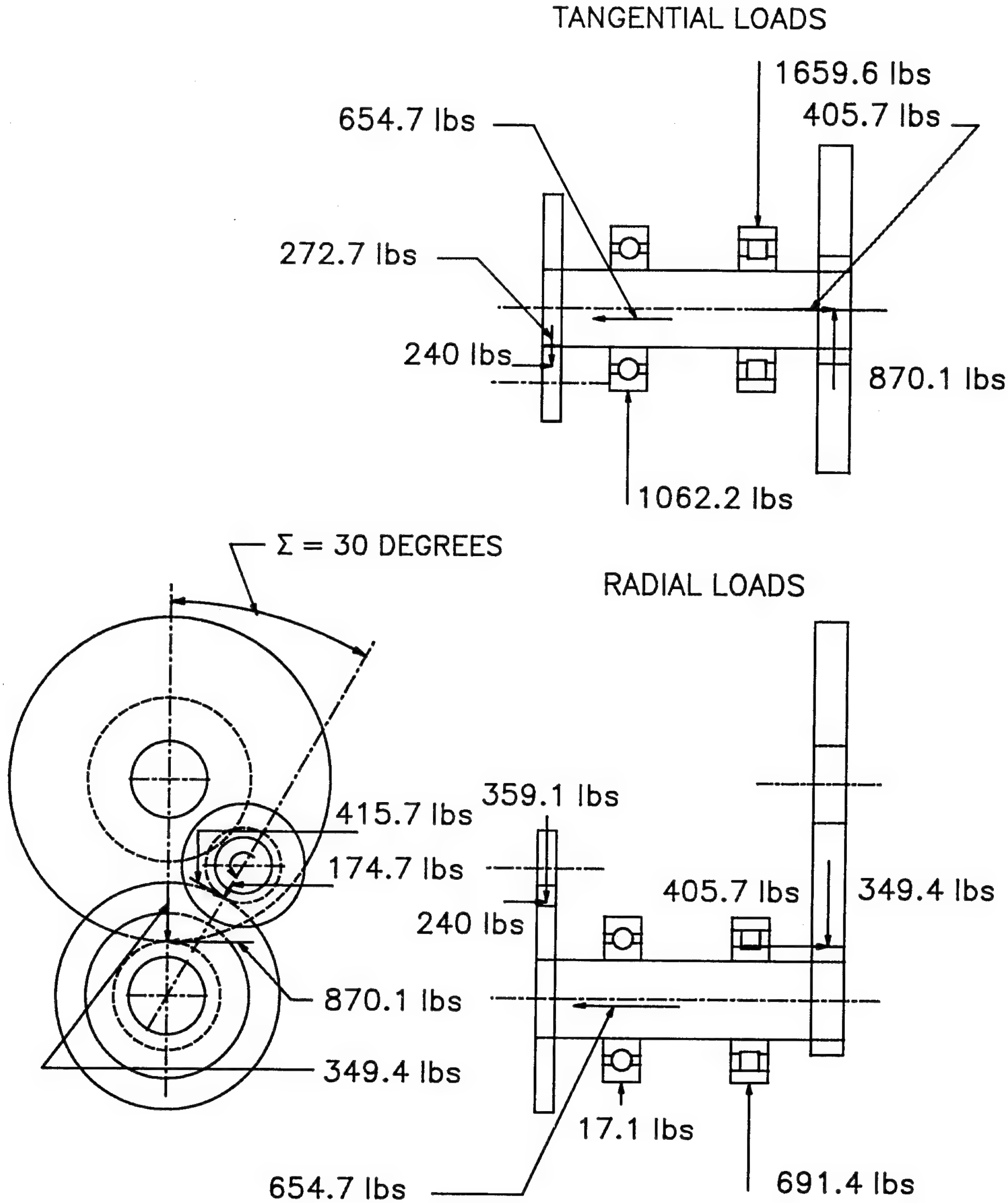


FIGURE 22 — INTERMEDIATE SHAFT LOADING FOR COMBINED HELICAL EXAMPLE

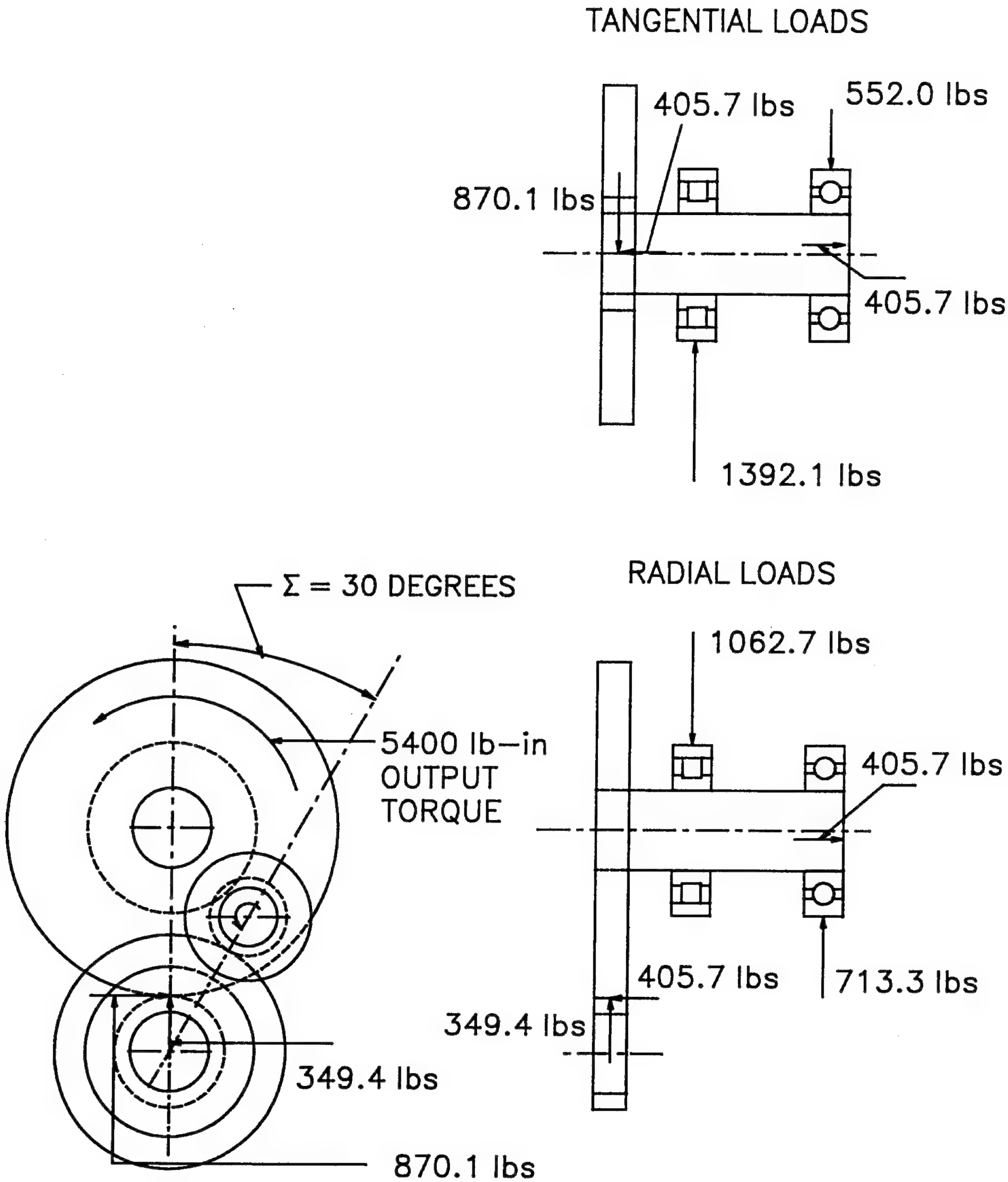


FIGURE 23 – OUTPUT GEAR SHAFT LOADING FOR COMBINED HELICAL EXAMPLE

Two-Stage Helical

The last application example for TLIFE is a two-stage, single-mesh helical-gear transmission. In this example, the transmission is composed of two single-mesh helical unit transmissions arranged in series. To demonstrate the flexibility of TLIFE and verify its life modeling, this two-stage single-mesh helical-gear transmission is matched to the compound helical-gear transmission of the prior example. A compound-mesh helical-gear transmission can be analyzed as a compound helical transmission or as two single-mesh, helical-gear transmissions in series. The two-stage single helical-gear transmission unit is shown in Figure 24.

This transmission has the same configuration, speed and torque loading as the compound helical reduction of the second example. As a result, most transmission properties used in the earlier example are also appropriate for this example. The transmission has an input torque of 600 lb-in in the counter-clockwise direction, and an input speed of 5,000 rpm. The transmission has an external output gear and a 30 degree shaft angle between the locations of the first and second unit transmission input to output center-lines.

The first unit transmission has a diametral pitch of 6, a 0.75 inch face width, a 20 degree normal pressure angle, a 30 degree helix angle, and a right-hand helix on the pinion. Its pinion has 15 teeth and its output gear has 45 teeth. Both gears have 0.167 inch addenda.

The second unit transmission has a diametral pitch of 4, a 20 degree normal pressure angle, an axial face width of 1.25 inches, and a 25 degree helix angle with a right-hand helix on the pinion. Its pinion has 12 teeth and its output gear has 45 teeth. Both gears have 0.25 inch addenda.

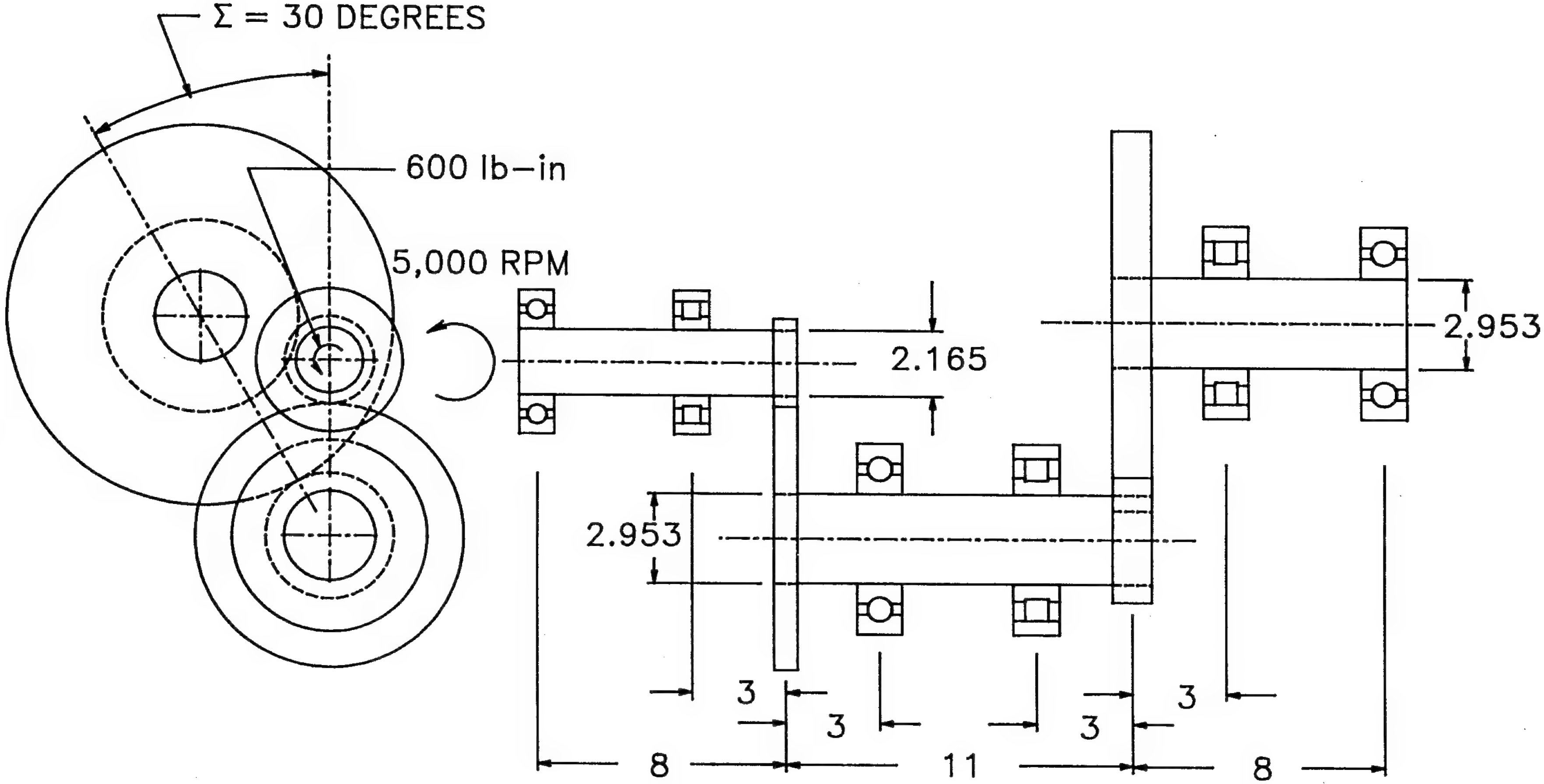


FIGURE 24 – TWO STAGE HELICAL EXAMPLE

All the gears are made of high strength steel with a surface material strength of 220 ksi. The Weibull slope for all four gears is 2.5, and the load-life exponent of the gears is 8.93.

Each gear is mounted in an overhung mounting with the ball bearing taking the thrust load. In each mounting, the first bearing is 3 inches from its gear and the second bearing is 8 inches from its gear. All the roller bearings have a Weibull slope of 9/8, all the ball bearings have a Weibull slope of 10/9 and all six bearings have a life adjustment factor of 6 and a load adjustment factor for race rotation of 1.

The two pinion bearings for the first unit transmission are 300 series with a 2.165 inch (55 mm) bore. The roller bearing is closest to the pinion and has a dynamic capacity of 4,070 lbs, while the ball bearing, which is furthest from the pinion, has a dynamic capacity of 2,248 lbs, a static capacity of 7,880 lbs, and a 25 degree contact angle.

The output gear of the first unit transmission is mounted in an overhung mounting with the first bearing being the ball bearing and the second bearing being the straight-row roller bearing. These bearings are 300 series with a 2.953 inch (75 mm) bore. The ball bearing has a dynamic capacity of 4,160 lbs, a static capacity of 13,100 lbs, and a 25 degree contact angle, while the roller bearing has a dynamic capacity of 8,810 lbs.

Both gears of the second unit transmission have overhung mountings with the straight-roller bearing closest to the gear and the ball bearing furthest from the gear. The second unit transmission bearings are 300 series with a 2.953 inch (75 mm) bore. The roller bearings have a dynamic capacity of 8,810 lbs, and the ball bearings have a dynamic capacity of 4,160 lbs, a static capacity of 13,100 lbs, and a 25 degree contact angle.

In this description, the shaft locations of the bearings are defined differently from the second example because there is no intermediate shaft in the two-stage single helical transmission. The intermediate shaft of the compound helical gear has two bearings. The first bearing is a ball bearing and the second is a roller bearing. These two bearings are identified as both the two bearings on the output shaft of transmission one and the two bearings on the input shaft of transmission two in this example. Thus, they must be identical in location and capacity. The output shaft of the first transmission has the ball bearing as its first bearing and the roller bearing as its second. However, the input shaft of the second transmission has the roller bearing as its first bearing and the ball bearing as its second bearing to accomplish this.

For the two-stage, single-mesh helical transmission, there are 23 lines of data in the input file. Thus, five additional lines of input data are required for the sequential single helical gear as opposed to the compound helical gear. A second unit number line and a unit configuration property line are required to describe the second transmission. Three additional lines are needed to describe the shaft geometry and properties and the bearing properties of the two bearings for the input shaft of the second transmission.

Table 9 summarizes the input data of the two-stage single-mesh helical transmission. Table 9 consists of 23 sections which represent the 23 lines of data required in the input file, which is listed in Table 10.

The first section lists the overall transmission system data, which now lists two unit transmissions. The next eleven sections describe the first unit transmission and the last eleven sections describe the second unit transmission. For each unit transmission, the first section gives the type of

TABLE 9

TWO-STAGE SINGLE HELICAL GEAR UNIT INPUT DATA

Overall Transmission

Number of unit transmissions	2
English system of units	2
System input torque	600.0 lb-in
System input speed	4500.0 RPM
Counter-clockwise input torque and rotation	1

Transmission 1

Unit Number

Single helical gear unit number	2

Unit Configuration Properties

External output gear	1

Gear Mesh Properties

Normal diametral pitch	6.0 1/inch
Normal pressure angle	20.0 degrees
Axial face width	0.75 inches
Helix angle	30.0 degrees
Right-hand pinion helix	1

Pinion Properties

Number of teeth on the pinion	15
Addendum	0.167 inch
Gear Weibull slope	2.5
Gear surface material strength	220.0 ksi
Gear surface load-life exponent	8.93

Pinion Support Shaft

Only bearing 2 supports the thrust load	2
Overhung mounting	2
Distance A	3.0 inches
Distance B	8.0 inches

Pinion Bearing 1

Single-row straight-roller bearing	3
Basic dynamic capacity	4070.0 lbs
Weibull slope	1.125
Life adjustment factor	6.0
Load adjustment factor for race rotation	1.0

TABLE 9 CONTINUED

Pinion Bearing 2
Single-row ball bearing 1
Basic dynamic capacity .... 2248.0 lbs
Weibull slope 1.1111
Life adjustment factor 6.0
Load adjustment factor for race rotation 1.0
Static capacity of ball bearing .... 7880.0 lbs
Bearing contact angle .... 25.0 degrees

Output Gear Properties
Number of teeth on the output gear 45
Addendum .... 0.167 inch
Gear Weibull slope 2.5
Gear surface material strength .... 220.0 ksi
Gear surface load-life exponent 8.93

Gear Support Shaft
Only bearing 1 supports the thrust load 1
Overhung mounting 2
Distance A .... 3.0 inches
Distance B .... 8.0 inches

Gear Bearing 1
Single-row ball bearing 1
Basic dynamic capacity .... 4160.0 lbs
Weibull slope 1.1111
Life adjustment factor 6.0
Load adjustment factor for race rotation 1.0
Static capacity of ball bearing .... 13100.0 lbs
Bearing contact angle .... 25.0 degrees

Gear Bearing 2
Single-row straight-roller bearing 3
Basic dynamic capacity .... 8810.0 lbs
Weibull slope 1.125
Life adjustment factor 6.0
Load adjustment factor for race rotation 1.0

Transmission 2

Unit Number
Single helical gear unit number 2

Unit Configuration Properties
External output gear 2
Shaft angle .... 30.0 degrees

TABLE 9 CONTINUED

<u>Gear Mesh Properties</u>	
Normal diametral pitch	4.0 1/inch
Normal pressure angle	20.0 degrees
Axial face width	1.25 inches
Helix angle	25.0 degrees
Right-hand pinion helix	1
<u>Pinion Properties</u>	
Number of teeth on the pinion	15
Addendum	0.25 inches
Gear Weibull slope	2.5
Gear surface material strength	220.0 ksi
Gear surface load-life exponent	8.93
<u>Pinion Support Shaft</u>	
Only bearing 2 supports the thrust load	2
Overhung mounting	2
Distance A	3.0 inches
Distance B	8.0 inches
<u>Pinion Bearing 1</u>	
Single-row straight-roller bearing	3
Basic dynamic capacity	8810.0 lbs
Weibull slope	1.125
Life adjustment factor	6.0
Load adjustment factor for race rotation	1.0
<u>Pinion Bearing 2</u>	
Single-row ball bearing	1
Basic dynamic capacity	4160.0 lbs
Weibull slope	1.1111
Life adjustment factor	6.0
Load adjustment factor for race rotation	1.0
Static capacity of ball bearing	13100.0 lbs
Bearing contact angle	25.0 degrees
<u>Output Gear Properties</u>	
Number of teeth on the output gear	45
Addendum	0.25 inches
Gear Weibull slope	2.5
Gear surface material strength	220.0 ksi
Gear surface load-life exponent	8.93
<u>Gear Support Shaft</u>	
Only bearing 2 supports the thrust load	2
Overhung mounting	2
Distance A	3.0 inches
Distance B	8.0 inches

TABLE 9 CONTINUED

Gear Bearing 1
Single-row straight-roller bearing 3
Basic dynamic capacity .... 12070.0 lbs
Weibull slope 1.125
Life adjustment factor 6.0
Load adjustment factor for race rotation 1.0

Gear Bearing 2
Single-row ball bearing 1
Basic dynamic capacity .... 5690.0 lbs
Weibull slope 1.1111
Life adjustment factor 6.0
Load adjustment factor for race rotation 1.0
Static capacity of ball bearing .... 13100.0 lbs
Bearing contact angle .... 25.0 degrees

TABLE 10

TWO-STAGE SINGLE HELICAL GEAR UNIT INPUT DATA FILE

2	2	600.0	5000.0	1		
2						
1						
6.	20.	0.75	30.0	1		
15	0.167	2.5	220.0	8.93		
2	2	3.	8.			
3	4070.	1.125	6.	1.0		
1	2248.	1.1111	6.	1.0	7880.0	25
45	0.167	2.5	220.0	8.93		
1	2	3.	8.			
1	4160.	1.1111	6.	1.0	13100.0	25
3	8810.	1.125	6.	1.0		
2						
1	30.0					
4.	20.	1.25	25.0	1		
15	0.25	2.5	220.0	8.93		
2	2	3.	8.			
3	8810.	1.125	6.	1.0		
1	4160.	1.1111	6.	1.0	13100.0	25
45	0.25	2.5	220.0	8.93		
2	2	3.	8.			
3	12070.	1.125	6.	1.0		
1	5690.	1.1111	6.	1.0	13100.0	25

unit transmission, the second section gives the unit configuration properties which includes just the type of output gear for the first unit and both the type of output gear and the shaft angle between the first and second unit transmissions for the second unit. Following these two sections, the third section gives the gear mesh properties, the fourth section describes the pinion, the fifth section describes the pinion mounting geometry, the sixth and seventh give the pinion bearing data, the eighth gives the output gear properties, the ninth describes the output shaft mounting and the tenth and eleventh describe the output gear bearings.

Once the input file has been created, TLIFE can be run by typing TLIFE when in a directory which contains "TLIFE.EXE," as described earlier. Appendix D lists the output file for this example.

Table 11, like Table 7, summarizes the axial, radial, and tangential forces on the gears and bearings which are reported in the output file. Most of the component values in these two tables are identical; however, note that the bearing order of the roller and ball bearings on the input shaft of the second transmission is in the reverse order of the bearings on the intermediate shaft of the compound helical gear, to match the transmission descriptions. Thus, when comparing the two tables, the roller bearing in one table corresponds to the roller bearing in the other table. A similar comparison exists for the ball bearings in the two tables. Another difference in notation between the two tables is that for the compound helical-gear transmission, the first intermediate gear corresponds to the output gear of the first unit transmission of the two-stage single-mesh helical transmission and the second intermediate gear corresponds to the input pinion of the second unit transmission. The loading conditions on the gears and bearings for the

TABLE 11

TWO-STAGE SINGLE HELICAL GEAR UNIT COMPONENT FORCES

COMPONENT	AXIAL FORCE (lbs)	RADIAL FORCE (lbs)	TANGENTIAL FORCE (lbs)
Transmission 1			
Input Pinion	240.0	174.7	415.7
Input Roller Bearing	0.	210.2	665.1
Input Ball Bearing	240.0	-35.5	-249.4
Output Gear	-240.0	174.7	415.7
Transmission 2			
Input Pinion	-405.7	349.4	870.1
Input Roller Bearing	0.	691.4	1,659.6
Input Ball Bearing	-645.7	17.1	1,062.2
Output Gear	-405.7	349.4	870.1
Output Roller Bearing	0.	1,062.7	1,392.1
Output Ball Bearing	-405.7	-713.3	-522.0

TABLE 12

TWO-STAGE SINGLE HELICAL GEAR UNIT CAPACITIES AND LIVES

COMPONENT	DYNAMIC CAPACITY (lb-in)	L10 LIFE IN MILLION OUTPUT ROTATIONS	MEAN LIFE (hours)
Transmission 1			
Input Pinion	34,089	13,995,620	941,329,300
Input Roller Bearing	27,865	225	44,242
Input Ball Bearing	33,979	249	50,267
Output Gear	36,701	27,056,090	1,819,761,000
Transmission	24,795	127	23,530
Transmission 2			
Input Pinion	39,218	48,929,250	3,290,924,000
Input Roller Bearing	32,646	379	74,610
Intput Ball Bearing	26,642	120	24,231
Output Gear	42,223	94,589,140	6,361,955,000
Output Roller Bearing	64,052	3,505	689,796
Output Ball Bearing	63,167	1,601	322,951
Transmission	23,406	92	16,886
Combined Overall Transmission	19,753	57	11,426
Components			9,831

two-stage single-mesh helical-gear transmission are identical to those for the compound helical-gear transmission. Thus, Figures 19, 20, and 21 represent the loading conditions on the gears and bearings for the input and output shafts of the two-stage single-mesh helical-gear transmission as well.

Table 12, similar to Table 8, summarizes the primary output of the TLIFE program for this example. To compare the results in Table 8 and Table 12, it is necessary to compare the values of the input roller bearing and input ball bearing of the second transmission in Table 12 with the intermediate roller bearing and intermediate ball bearing in Table 8, respectively. All the other component values in Table 12 can be compared directly to the corresponding component values for the compound helical-gear transmission in Table 8.

Upon comparison of the two tables, one sees that the values for the dynamic capacities and the mean lives of the gears, bearings, and the unit transmission are identical. The identical values in the load and in the capacity and lives tables, and in the input and loading figures for the compound helical gear and two-stage single-mesh helical-gear transmissions show the flexibility of TLIFE's modeling capability.

PROGRAM STRUCTURE

The structure of the computer program is shown in the block diagram of Figure 25. A main program opens the input and output ASCII files, reads the number and type of transmissions to be analyzed and calls one of three configuration routines to analyze the selected transmission. SMESH analyzes the single-mesh unit transmissions, COMRED analyzes the compound transmissions and PLANET analyzes the planetary and star configurations.

The program analyzes a single transmission or a series of connected transmissions based on the data contained in the input file. Each configuration analysis routine reads in the data from the input file which describes the unit transmission, performs a motion and load analysis of the unit transmission and then analyzes the lives and dynamic capacities of the components in the unit transmission. The main program then calls the life and dynamic capacity analysis routines for each unit transmission, and after all the unit transmissions have been analyzed, superimposes the loading on the inter-connecting bearings, analyzes the overall transmission for its system life and capacity, and calls the output routine to record the results in an output file. Once this transmission analysis is completed, the program closes the input and output files and prompts the user to determine if another transmission is to be analyzed. If so, the program repeats its analysis for another input data file. If not, the program terminates.

In the program, a common block array, PROP, serves as the property database. This array abstracts the component and transmission property values from the analysis routines. It is a two dimensional array, with the rows corresponding to specific components of a transmission and the columns containing values of specific properties. The first column of the array

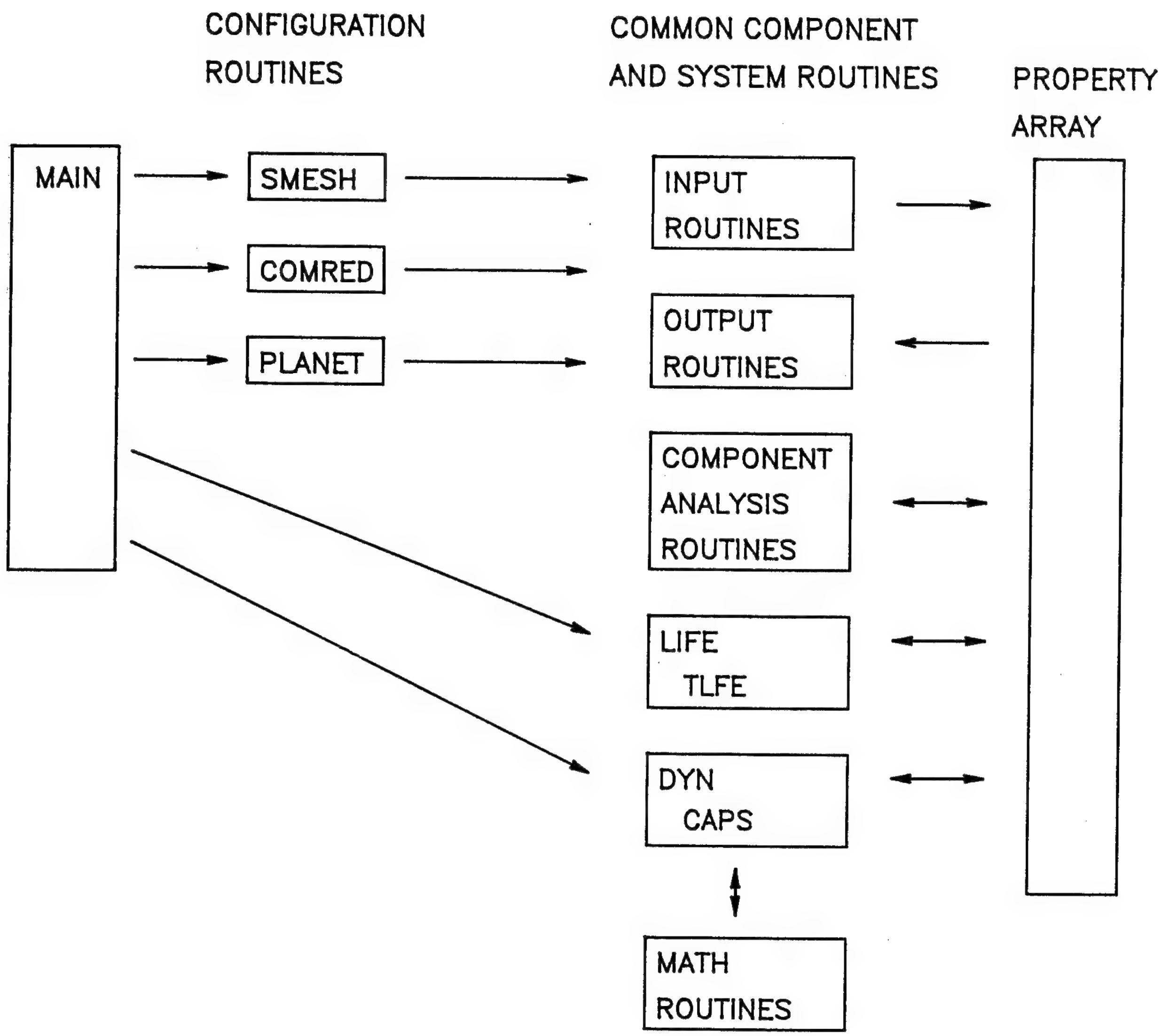


FIGURE 25 – TLIFE PROGRAM FLOW CHART

contains the system properties for the entire transmission, and the first column for each unit transmission contains the system properties for that unit.

Since the routines that evaluate the transmission life and dynamic capacity interface solely with the property array, some elements of the property array are independent of any specific transmission configuration. The system analysis routines are structured in an identical manner for all configurations considered. Thus, other configurations can be added to the program by adding appropriate configuration analysis routines along with any supporting routines used by these configurations.

As shown in Figure 25, the program is modular in structure: the configuration specific routines use the same input and output routines, component analysis routines, basic mathematics routines and system analysis routines.

Since the subroutines that determine the transmission life and dynamic capacity interface directly with the property array, they are separated from any specific transmission configuration. The system analysis subroutines, LIFE and DYN, work in an identical manner for all unit transmission configurations. For the full system, the same routines are used with a pointer array to include all the components in the overall transmission while skipping the unit transmission properties. Thus, other configurations can be added to the program by adding appropriate configuration analysis subroutines.

When analyzing a series transmission without a dual spiral-bevel transmission, the data file has the data for all unit transmissions ordered from input to output, and the main program calls the particular configuration specific routines one after another in the same order. After analyzing all

the unit transmissions in the overall transmission, the analysis of the overall transmission is performed in the main program by calling the life and system dynamic capacity analysis routines, and writing the results to the output file.

When a series transmission includes a dual spiral-bevel transmission as one of the unit transmissions, the data in the input file for the analysis is ordered from input to output for one input branch of the dual spiral-bevel transmission, followed by data for the dual spiral-bevel transmission and then by data for the unit transmissions in order from the output of the spiral-bevel transmission to the final output of the overall transmission. The unit transmissions are analyzed in the same order, the overall transmission analysis is performed in the main program where the input branch elements are counted twice and the results are written to the output file from the main program.

The right and left branch of the dual spiral-bevel transmission may be comprised of any of the remaining unit transmissions. This restricts the maximum number of inputs in the overall transmission to two. The number of unit transmissions in both branches can be zero, but they must be equal. Both the right branch and the left branch are assumed to be identical in all respects including the load levels so the data has to be provided only for one branch.

Table 13 is a list of the program subroutines with the starting line number of the subroutine in brackets and a brief description of the subroutine function. The subroutines are grouped by their function in the program. The first three are the configuration specific analysis routines. The next group are the input routines which read the data from the input file. These

TABLE 13

TLIFE SUBROUTINE LIST

Line Number	Routine	Description
[377]	MAIN PROGRAM	
[837]	SUBROUTINE SMESH	Configuration routines
[1091]	SUBROUTINE COMRED	
[1468]	SUBROUTINE PLANET	
[1935]	SUBROUTINE CASIP	Input routines
[2018]	SUBROUTINE BEARI	
[2099]	SUBROUTINE GEARI	
[2167]	SUBROUTINE MESHI	
[2265]	SUBROUTINE TSYSIN	
[2340]	SUBROUTINE SYSIN	
[2522]	SUBROUTINE BEARO	Output routines
[2578]	SUBROUTINE GEARSO	
[2754]	SUBROUTINE GRINTO	
[2798]	SUBROUTINE MNTSO	
[2871]	SUBROUTINE MTINTO	
[2959]	SUBROUTINE OUTPUT	
[3313]	SUBROUTINE GEARGM	Gear mesh geometry calculations
[3579]	SUBROUTINE LOCATE	
[3621]	SUBROUTINE TESTCL	Assembly and thrust capability checks
[3679]	SUBROUTINE TESTDI	
[3723]	SUBROUTINE TESTDB	
[3789]	SUBROUTINE BRTHCK	
[3831]	SUBROUTINE LOAD	Gear and bearing load analyses
[3918]	SUBROUTINE BLC1	
[4028]	SUBROUTINE BLC2	
[4155]	SUBROUTINE SET	Gear life and capacity
[4207]	FUNCTION BSCAP	
[4240]	SUBROUTINE BDCAP	Bearing life and capacity
[4303]	SUBROUTINE BALL	
[4462]	SUBROUTINE TAPER	
[4503]	SUBROUTINE LIFE	System life and dynamic capacity
[4650]	SUBROUTINE TLFE	
[4674]	SUBROUTINE DYN	
[4733]	SUBROUTINE CAPS	
[4757]	SUBROUTINE GAMMA	Supporting mathematical calculations
[4800]	SUBROUTINE HALVE	
[4861]	SUBROUTINE LESQR	
[4918]	SUBROUTINE DINT	
[4984]	SUBROUTINE SINT	
[5032]	SUBROUTINE LINT	
[5060]	FUNCTION DEG	
[5066]	FUNCTION RAD	
[5072]	SUBROUTINE LOWER	Lower case generation for UNIX use
[5101]	END	

routines are matched to the components, independent of the configurations. The next set are output routines which are matched to the components and to the configurations. Following these are the component analysis routines. These determine: the gear mesh geometry for the spiral-bevel gears and the contact ratios for all gears, the loads on the gears and bearings, assembly interference possibilities, and the lives and dynamic capacities of the components. Next are the routines LIFE and TLFE which perform the system life analysis for all configurations and the overall transmission when analyzing a series of transmissions by interacting only with the property array. These are followed by the subroutine DYN and its supporting routine CAPS which perform the system dynamic capacity analysis similar to the life analysis using the property array. Following these are some basic mathematics routines which support the analysis routines. These are followed by the routine LOWER, which is used to convert the input file name to lower case for consistency with lower case filenames in the UNIX environment.

Table 14 lists the metric and English unit property arrays showing the elements of the property array for the overall transmission, the unit transmissions, and the gears and bearings along with their units. The array has a variable MET, as the eighteenth element of the first column, which is the metric - English unit flag. This flag has a value of one for metric system analysis and two for the English system analysis. It alerts the program to the difference between the metric module and the English diametral pitch in the gear size definition. It also selects the unit conversion factor for system power calculations and the units for the variables in the output file. Metric inputs produce metric outputs and the English inputs produce English outputs with the same analysis algorithm throughout the program.

TABLE 14

PROPERTY ARRAY PROP(260,35)

item	Entire Transmission 1	Unit Transmission 2,...
1	ℓ_{av} Life (hours)	ℓ_{av} Life (hours)
2	$\ell_{av,c}$ Comp. Life (hrs)	$\ell_{av,c}$ Comp. Life (hrs)
3	ℓ_{10} Life (M Cyl Out)	ℓ_{10} Life (M Cyl Out)
4	ℓ_{10} Life (hours)	ℓ_{10} Life (hours)
5	p_s - load-life exp	p_s - load-life exp
6	b_s - Weibull exp	b_s - Weibull exp
7	D_s - Dyn Cap (kN-m) / (lb-in)	D_s - Dyn Cap (kN-m) / (lb-in)
8	ω_{in} (RPM)	ω_{in} (RPM)
9	ω_{out} (RPM)	ω_{out} (RPM)
10	T_{out} (kN-m) / (lb-in)	T_{out} (kN-m) / (lb-in)
11	n - reduction ratio	n - unit reduction ratio
12		Σ (degrees)
13		Σ (radians)
14	Power (kW) / (HP)	Power (kW) / (HP)
15		NP - no. of planets
16		NRING
17		NARM
18	MET - metric / English switch	Λ (degrees)
19		Λ (radians)
20		β (degrees)
21	T_{in} (kN-m) / (lb-in)	T_{in} (kN-m) / (lb-in)
22		IGEAR - gear type
23		JOIN - first or addnl. unit
24	IDIRN - input direction	IDIRN - input direction
25	IDIRO - output direction	IDIRO - output direction
26		NOUT - output location
27		Σ_i (degrees)
28		Σ_i (radians)
29		Σ_o (degrees)
30		Σ_o (radians)

TABLE 14 CONTINUED

PROPERTY ARRAY PROP(260,35)

item	Gears	Bearings
	3,...	...
1	ℓ_{av} Life (hours)	ℓ_{av} Life (hours)
2	ℓ_{10} Life (M Cyl)	ℓ_{10} Life (M Cyl)
3	ℓ_{10} Life (M Cyl Out)	ℓ_{10} Life (M Cyl Out)
4	ℓ_{10} Life (hours)	ℓ_{10} Life (hours)
5	pg - load-life exp	p_b - load-life exp
6	b_g - Weibull exp	b_b - Weibull exp
7	D_g (kN-m) / (lb-in)	D_b (kN-m) / (lb-in)
8	C_g -for M Cyl Out (kN)/(lbs)	C_b - for M Cyl Out (kN) / (lbs)
9	ω (RPM)	ω (RPM)
10	T_g (kN-m) / (lb-in)	ITY - bearing type
11	N_g - no. of teeth	C_o - static capacity (kN) / (lbs)
12	ϕ (degrees)	F_{oa} - axial preload (kN) / (lbs)
13	ϕ (radians)	α - contact (degrees)
14	module (mm) / P_d (in^{-1})	F_r / F_a rating factor
15	R_g (mm) / (in)	V - Race Rotation Factor
16	F_n - normal (kN) / (lbs)	F_{re} - eq. radial (kN) / (lbs)
17	F_r - radial (kN) / (lbs)	F_r - gear radial dir (kN) / (lbs)
18	F_t - tan (kN) / (lbs)	F_t - gear tan dir (kN) / (lbs)
19	f (mm) / (in)	F_{total} - radial (kN) / (lbs)
20	S_{ac} (MPa) / (ksi)	A, B, C or E dist (mm) / (in)
21	ℓ_c load cycles per rev	a - life factor
22	ρ curv radius (mm) / (in)	X Radial Factor

TABLE 14 CONTINUED

PROPERTY ARRAY PROP(260,35)

item	Gears	Bearings
	3,...	...
23	Σ for bevels	Y Thrust Factor
24	C_g for M Cyl (kN) / (lbs)	C - base dyn cap (kN) / (lbs)
25	F_a - axial (kN) / (lbs)	F_a - axial (kN) / (lbs)
26	A_o (mm) / (in)	ITHR - thrust switch
27	ψ (degrees)	ICS - mounting switch
28	IHAND - helix/spiral dir	
29	Γ_g (degrees)	
30	Addendum (mm) / (in)	D - between gears (mm) / (in)
31	R_e (mm) / (in)	
32	m_c - involute contact ratio	
33	m_a - face contact ratio	
34	f_e (mm) / (in)	
35	IDIRN - direction of ω	

When analyzing a series of transmissions, all unit transmissions must have their input in the same system, as the flag is set once in data group A for the entire transmission. Also the input torque, input speed and input rotation direction are given for the entire transmission along with the number of unit transmissions on this data line. For transmissions which include the dual-bevel unit, the number of transmissions is the number in one branch from system input to system output.

In this array, the elements include common values across the same row for all unit transmissions to prepare for the system analysis. Row 1 contains the output mean life of the components and transmission in hours. Row 2 contains the component mean life in million component load cycles. Row 3 contains the ninety-percent reliability life of the system and components in million output rotations, with Row 4 containing the same life in hours. Row 5 contains the load-life exponent for each component and unit transmission. Row 6 contains the Weibull slope for each component and unit transmission. And Row 7 contains the dynamic capacities of all of these elements in units of output torque. Higher numbered rows contain properties which vary from element type to element type. The table lists all the elements of the property array.

As properties of all components are required for calculating the life and dynamic capacity of the overall transmission, the property array must have these properties for all components before the system analysis is performed. Once the analysis of all unit transmissions is done, the analysis of the overall transmission is performed in the main program and the results written to the output file.

Both the unit transmission and the full system analyses are performed with the same system life and capacity routines. The routines sum functions of elements four through seven from the property array columns for the transmissions bearings and gears. By using a pointer array which can be different for each unit transmission and for the entire transmission, the property array and the system life and capacity routines can remain fixed for all configurations. The pointer array allows the unit transmission property columns to keep their place in the property array immediately ahead of their components, yet not be counted when the full set of components is counted for the full system analysis. Additionally, it enables the life calculation of a two-branch transmission with only one branch present in the property array by using the appropriate columns twice in the analysis.

The maximum number of unit transmissions in one branch of the overall transmission that can be analyzed is at present set to 25, and the maximum number of components in the overall transmission cannot be more than 259 less the number of unit transmissions, which is one less than the size of the property array. To analyze a transmission composed of more than 25 unit transmissions or more than 259 components, the dimensions of the appropriate arrays must be changed in the program.

The program takes the data from the input file and writes the results to an output data file. The input file includes: the number of unit transmissions, the selection of the SI-metric or inch-English system of units, the input shaft torque, speed and direction. For each unit transmission, it also includes: descriptions of the unit geometry and component gears and bearings and their dynamic capacities. The details for creating the input file may be found in the section PROGRAM USE of this report, in the file

"TLIFE.DOC" and at the beginning of the program source code, "TLIFE.FOR." The file "TLIFE.DOC" is included as Appendix A.

The output data file includes a repetition of the input data to define the transmission, the power, a report of the component loads and capacities, a summary of the dynamic capacities in units of output torque, ninety-percent reliability lives in million output rotations and hours and the mean lives in hours for all components and the transmission. Following this are the mean life of the transmission and the overall mean component life in hours. This is repeated for all individual transmissions in the order of analysis.

Concluding the output are the mean life and the overall mean component life in hours for the overall transmission. The transmission mean life predicts the mean time between service overhauls for maintenance by full transmission replacement. In contrast, the overall mean component life predicts the mean time between service overhauls for maintenance by failed component replacement.

GEOMETRY, MOTION ANALYSIS AND COMPONENT LOAD CYCLES

For this analysis, the unit transmissions are treated sequentially to determine their gear geometry, speed reduction ratio, and component load cycles. Component load cycles are counted directly and in output rotations. The speed of the output shaft, which is the lowest in the entire transmission considering the fact that the transmissions analyzed are reduction units, is used as a common counting base and the dynamic capacities are given in units of output torque. The motion analysis for each unit transmission is performed in the respective analysis routine. In analyzing combined units, the output speed, torque and direction of rotation of a unit is passed to the next unit as its input. It is assumed that there are no transmission losses between the units.

Single Mesh Reductions

For the single-mesh spur and helical-reduction units of cases 1 and 2, the geometry is defined by: the numbers of teeth on the pinion and on the gear, the addenda of the pinion and the gear, the module or diametral pitch of the mesh normal to the teeth, the helix angle and its hand on the pinion, and whether the output gear is an internal gear or an external gear. The pitch diameter of the gear and pinion are calculated using the number of teeth, N, the normal module, m, or normal diametral pitch, P_d, and the helix angle, ψ, depending on the system of units as:

$$D = \frac{N \cdot m}{\cos \psi} \tag{1}$$

in metric, and

$$D = \frac{N}{P_d \cdot \cos \psi} \tag{2}$$

in English units. For spur gears, the helix angle is zero and the relations still hold.

To treat the strength of the helical teeth in mesh, the helical gears are modeled as equivalent planar spur gears in the normal plane. For helical gears, the diameter of the equivalent spur gear is related to the actual pitch diameter as:

$$D_e = D / \cos^2 \psi \tag{3}$$

The face width of the equivalent spur gear is a function of the helical overlap in the gear, as developed in the AGMA Standards [17]. This relationship is a function of the length of the line of action in the transverse plane, Z; the axial pitch, p_x; and the base helix angle, ψ_b. As shown in Figure 26, the length of the line of action is a function of the addenda of the pinion and the gear.

$$\begin{aligned} Z = \; & [(D_p/2 + a_p)^2 - (D_p/2)^2\cos^2\phi]^{1/2} \\ & + [(D_g/2 + a_g)^2 - (D_g/2)^2\cos^2\phi]^{1/2} \\ & - (D_p/2 + D_g/2) \sin \phi \end{aligned} \tag{4}$$

The axial pitch is given by:

$$p_x = \frac{\pi \, m}{\sin \psi} \tag{5}$$

in metric units or by:

$$p_x = \frac{\pi}{P_d \sin \psi} \tag{6}$$

in English units. And the base helix angle is defined with:

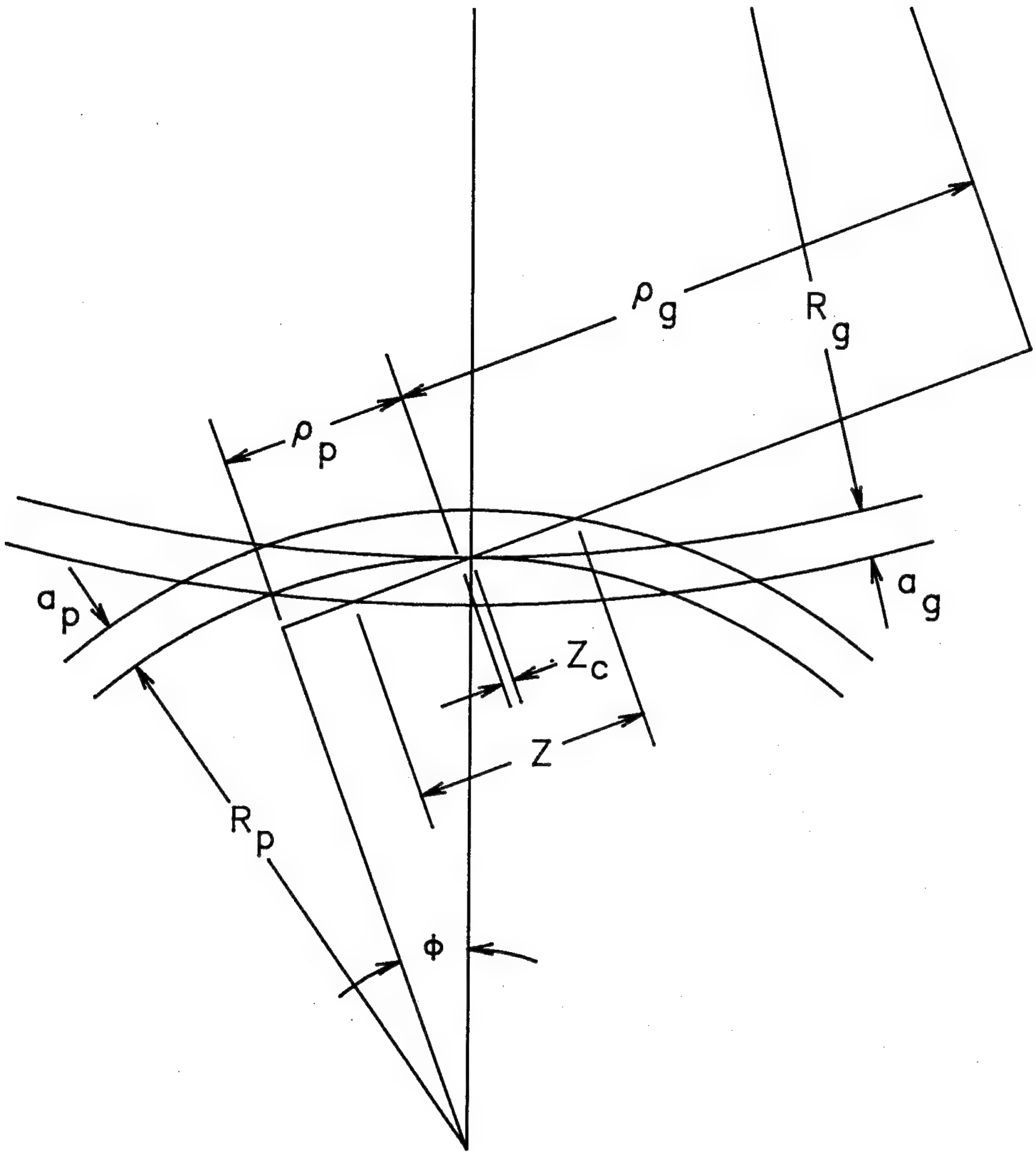


FIGURE 26 — GEAR—MESH LINE OF ACTION

$$\tan \psi_b = \tan \psi \cos \phi_n \tag{7}$$

Two types of helical overlap can occur: 1) full overlap with the axial contact ratio greater than one, and 2) partial overlap with the axial contact ratio less than one. With full helical overlap, there are always more than one tooth pair in mesh in the contact region. The axial contact ratio is:

$$m_a = \frac{f}{p_x} \tag{8}$$

For full helical overlap, the minimum-length sum of the lines of contact in the plane of action can be expressed as:

$$f_e = \frac{\mathrm{INT}(m_a)\ Z}{\sin \psi_b} + \frac{f - \mathrm{INT}(m_a)\ p_x}{\cos \psi_b} \tag{9}$$

where $\mathrm{INT}(m_a)$ is the integer portion of the axial contact ratio. Figure 27 shows the traces of the contacting tooth surfaces in the plane of action for a set of helical gears with full helical overlap. The minimum-length sum is the effective face width of the helical gear teeth. When the helical overlap is partial, the effective face width is given by a linear interpolation between the full overlap face width and the face width of a spur gear:

$$f_e = f \left(1.0 + m_a \left(\frac{Z}{p_x \sin \psi_b} - 1.0 \right) \right) \tag{10}$$

For full helical overlap, the tooth curvatures are taken in the center of the teeth. For the pinion:

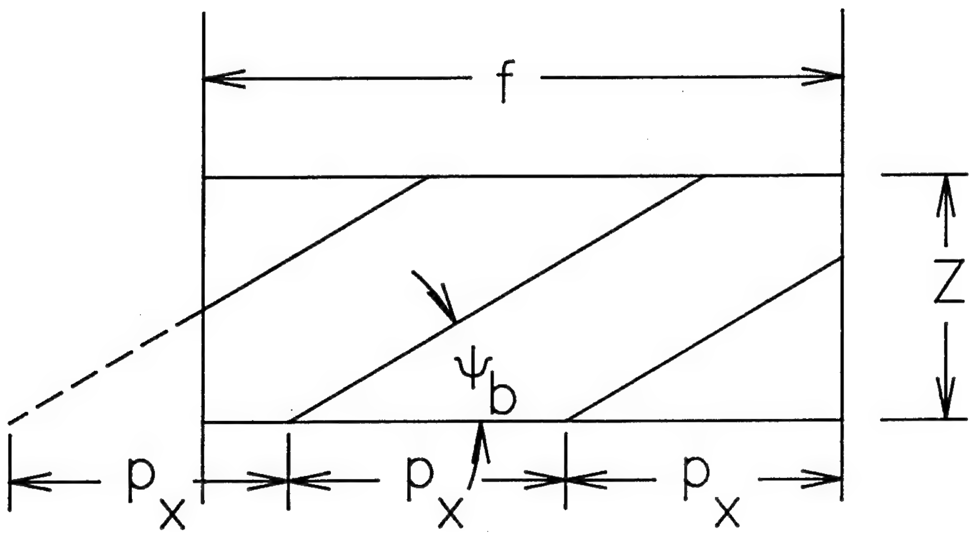


FIGURE 27 – HELICAL CONTACT LINES
IN THE PLANE OF ACTION

$$\rho_p = \frac{D_{pe}}{2} \sin \phi_n - Z_c \tag{11}$$

where Z_c is a small correction distance along the line of action from the pitch point to the mean pinion radius. The radius of curvature of the gear tooth is then:

$$\rho_g = \frac{D_{pe} + D_{ge}}{2} \sin \phi_n - \rho_p \tag{12}$$

For partial helical overlap and for spur gears, the pitting stress is calculated at the lowest point of single tooth contact on the pinion:

$$\rho_p = \left(\left(\frac{D_{pe}}{2} + a_p \right)^2 - \left(\frac{D_{pe}}{2} \right)^2 \cos^2 \phi \right)^{1/2} - \frac{\pi}{P_d} \cos \phi \tag{13}$$

and the corresponding radius of curvature of the gear tooth can be calculated with equation (12).

As the pitch diameters of the gears are independent of whether the gear is an external gear or an internal gear, the same relations for diameter (1) through (3) are used in both cases.

The gear ratio n, for the reduction is given as:

$$n = - R_o/R_i \tag{14}$$

where R_o is negative for an internal gear.

The output speed is the input speed divided by the gear ratio and each component sees a single load cycle per rotation. The input pinion and the bearings supporting it rotate at the input speed and the bearings supporting the output gear rotate at the output speed. From a standard gear train analysis, the ratio of the number of load cycles of the output gear for the input pinion is given as R_o/R_i.

Compound Reductions

For the compound spur and helical-reduction units of cases 3 and 4, the number of teeth on each gear, the addenda of the gears, the normal module or diametral pitch of each of the meshes, the helix angle and its hand on the pinion, the shaft angle, Σ, and whether the output gear is an internal gear or an external gear are required. The shaft angle, Σ, is measured from the input shaft and intermediate shaft center-line to the intermediate shaft and output shaft center-line, about the intermediate shaft axis counter-clockwise as seen from the input shaft. This angle is shown in Figure 6. The pitch diameters of all the gears in the meshes are found using equation (1) or (2) depending on the system of units.

The input and output gears are on different shafts separated by an intermediate shaft and each of the shafts rotate at different speeds. Thus there are three different speeds in the transmission. The two mesh reductions, n1 and n2 are:

$$n1 = -R_{ii}/R_i \tag{15}$$

and,

$$n2 = -R_o/R_{io} \tag{16}$$

where the overall transmission ratio is the product of these two reductions:

$$n = n1 \cdot n2 \tag{17}$$

Each component rotates at the speed of its respective shaft, and so the components see different load cycles. The relation between the load cycles of each of the components to the number of rotations of the output shaft is given in Table 15 with N_p, the number of parallel load paths, equal to 1. Table 15 gives the ratio of the load cycles of the components to the number of output

TABLE 15

LOAD CYCLE RATIOS

Component	Load Cycles
Input Pinion	$\dfrac{N_p \cdot (R_{ii} \cdot R_o)}{(R_i \cdot R_{io})}$
Intermediate Gear on Input Side	R_o/R_{io}
Intermediate Gear on Final Side	R_o/R_{io}
Final Gear	N_p

Where

R_i = radius of input pinion,

R_{ii} = radius of intermediate gear meshing with input pinion,

R_{io} = radius of intermediate gear meshing with final gear,

R_o = radius of final gear, and

N_p = number of parallel load paths.

shaft rotations for compound reductions, parallel compound reductions, and reverted reductions.

Parallel Compound Reductions

For the parallel compound reduction spur and helical units of cases 5 and 6, the number of teeth on each gear, the addenda of the gears, the normal module or diametral pitch of each of the meshes, the helix angle and its hand on the pinion, the input shaft angle, Σ_i, and whether the output gear is an internal gear or an external gear, and its location are required. Here the input shaft angle, Σ_i, which is shown in Figure 8, is measured between the center-lines of the two intermediate shafts with the input shaft about the input shaft. The motion analyses of these cases is nearly the same as that of the compound reductions. For this analysis, an output shaft angle, Σ_o, is calculated from the input shaft angle, Σ_i, and the size and location of the output gear and the intermediate gears which mesh with it. Figure 28 shows these shaft angles for external and internal output gears which are above and below the intermediate shafts. For output gears which are above the intermediate shafts, the output shaft angle is given by:

$$h = (R_i + R_{ii}) \cdot \sin(\Sigma_i/2) \tag{18}$$

$$\Sigma_o = 2.0 \cdot \sin^{-1} \left(\frac{h}{| R_o + R_{io} |} \right) \tag{19}$$

and the shaft angle, Σ, is calculated as:

$$\Sigma = \Sigma_i/2 - \Sigma_o/2 \tag{20}$$

For the output shaft located below the intermediate shafts, the shaft angle, Σ, is given as:

$$\Sigma = \pi + \Sigma_i/2 + \Sigma_o/2 \tag{21}$$

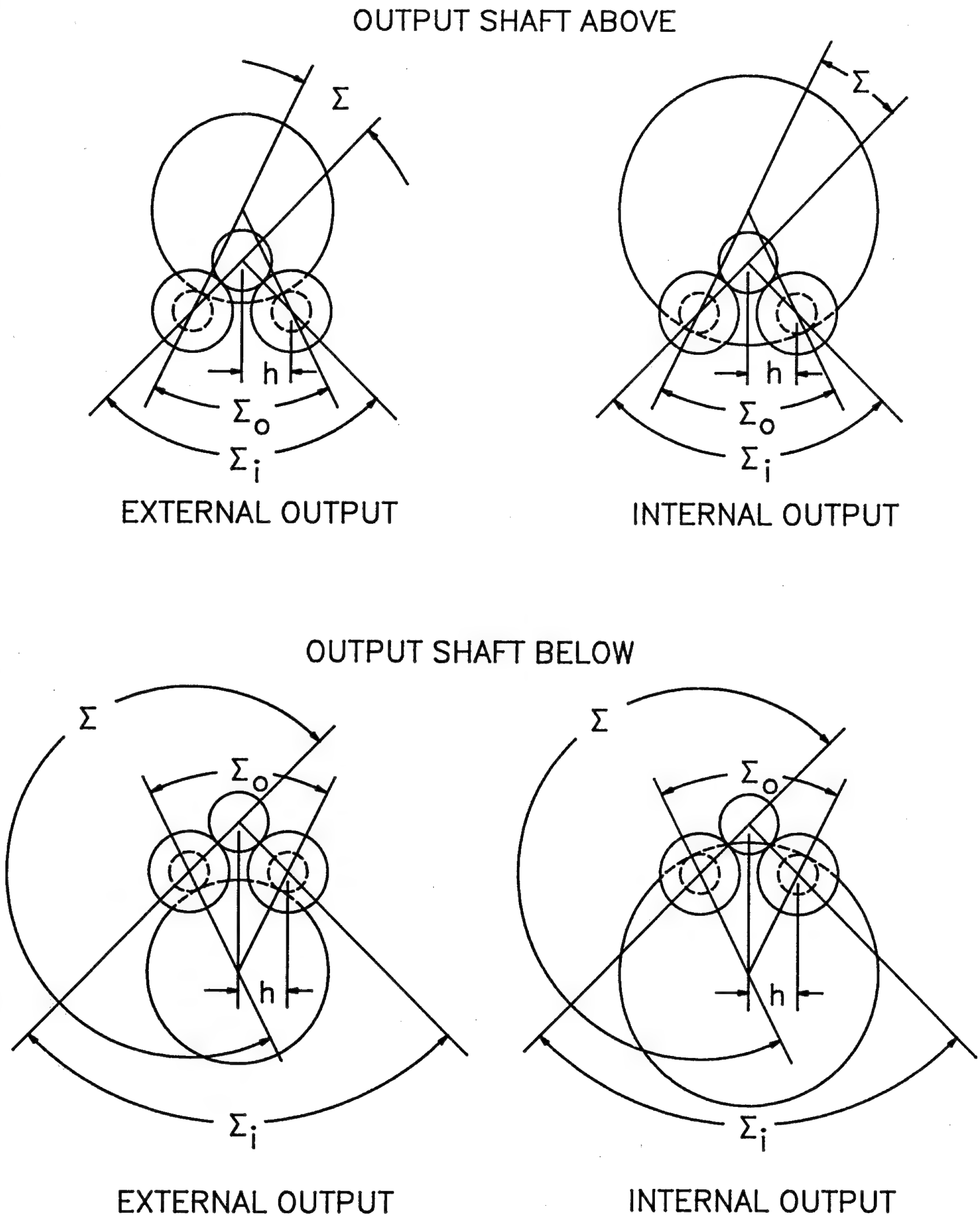


FIGURE 28 — PARALLEL—COMPOUND SHAFT ANGLES

The load cycle analysis differs from that of the compound reductions only in the value of N_p. The load cycle count for the input pinion and the output gear are twice those for the compound reductions. So, N_p is 2 in the equations of Table 15 for these cases.

Reverted Reductions

For the reverted reduction spur and helical units of cases 7 and 8, the number of teeth on each gear, the addenda of the gears, the normal module or diametral pitch of the meshes, the helix angle and its hand on the pinion, the number of parallel load paths, N_p, and whether the final gear is an internal or external gear are required. The pitch diameters of the gears in all the meshes are calculated using the relation (1) or (2) for each mesh.

As the output can come from either the final gear or the arm carrying the planets, the motions of the components can be different for the same physical unit. However, when the final gear is fixed and the arm is the output, the final gear still sees a number of load cycles as the planets rotate about it.

When the intermediate shafts and the arm holding them are fixed, the motion analysis of the reverted reductions is the same as for the compound reductions given in equations (1), (2) and (15) through (17).

With the final gear fixed, the reduction is a planetary with the output being the motion of the arm. From a planetary analysis approach, the speed of each gear, ω_j, is its speed relative to the arm, $\omega_{j/o}$, plus the arm speed, ω_o.

$$\omega_j = \omega_{j/o} + \omega_o \tag{22}$$

The speed of the planet gear with respect to the arm carrying the planets, $\omega_{p/o}$ is:

$$\omega_{p/o} = \frac{\omega_{i/o}}{n1} \tag{23}$$

And the speed of the final gear with respect to the arm carrying the planets, $\omega_{g/o}$ is:

$$\omega_{g/o} = \frac{\omega_{p/o}}{n2} = \frac{\omega_{i/o}}{n} \tag{24}$$

Expanding equation (22), noting that the speed of the final gear ω_g is zero for the fixed gear case, we have:

$$\omega_g - \omega_o = -\omega_o = \frac{\omega_i - \omega_o}{n} \tag{25}$$

or the speed of the output arm is:

$$\omega_o = \frac{\omega_i}{1 - n} \tag{26}$$

And, for the speed of the intermediate shaft or planets we have,

$$\omega_p - \omega_o = \frac{\omega_i - \omega_o}{n1} \tag{27}$$

or

$$\omega_p = \frac{\omega_i \cdot (n1 - n)}{n1 \cdot (1 - n)} \tag{28}$$

In these equations, the overall transmission ratio, n is positive for a reverted planetary reduction with a fixed external final gear and negative for a reverted planetary reduction with a fixed internal final gear. The first gear ratio, n1 is always negative and the second ratio, n2, has the opposite sign to that of the overall ratio, n.

The component load cycle counts for both the fixed-arm configuration and the planetary reduction are functions of the motion of the gears relative to the arm, so the load cycle counts for the star and planetary configurations

are the same. Since the direction of motion is not relevant, the cycle-count ratios to the output motion are as given in Table 15, with N_p being the number of intermediate shafts.

<u>Single-Plane Reductions</u>

For the single-plane reduction unit of case 9, the geometry is defined by whether the unit is stepped or non-stepped, the number of planets carried by the arm and whether the ring gear is the output gear or the arm is the output. The stepped reduction is shown in Figure 29. For this reduction unit, the sun gear is the input gear and the output is either from the ring gear or from the planet arm. In the case of the stepped single-plane reduction unit, the numbers of teeth on the sun gear, planet-sun gear, planet-ring gear and ring gear, the addenda of the gears and the modules or diametral pitches of the sun-planet mesh and of the planet-ring gear mesh are required. In the case of the non-stepped single-plane reduction unit, the numbers of teeth on the sun gear, planet gear, and ring gear, the addenda of the gears and the single module or diametral pitch of the meshes are required. In the case of a stepped single-plane reduction unit, the planet-sun gear is the planet gear meshing with the sun gear and the planet gear meshing with the ring gear is the planet-gear.

The pitch diameters of all the gears in the reduction unit are calculated using the relation (1) or (2) based on the system of units, with a helix angle of zero. This transmission also has three speeds and the two reduction ratios and overall transmission ratio are given by equations (15) through (17).

The motion and load cycle analyses for this case are the same as those for the reverted reduction with a ring gear output. The motion analysis is

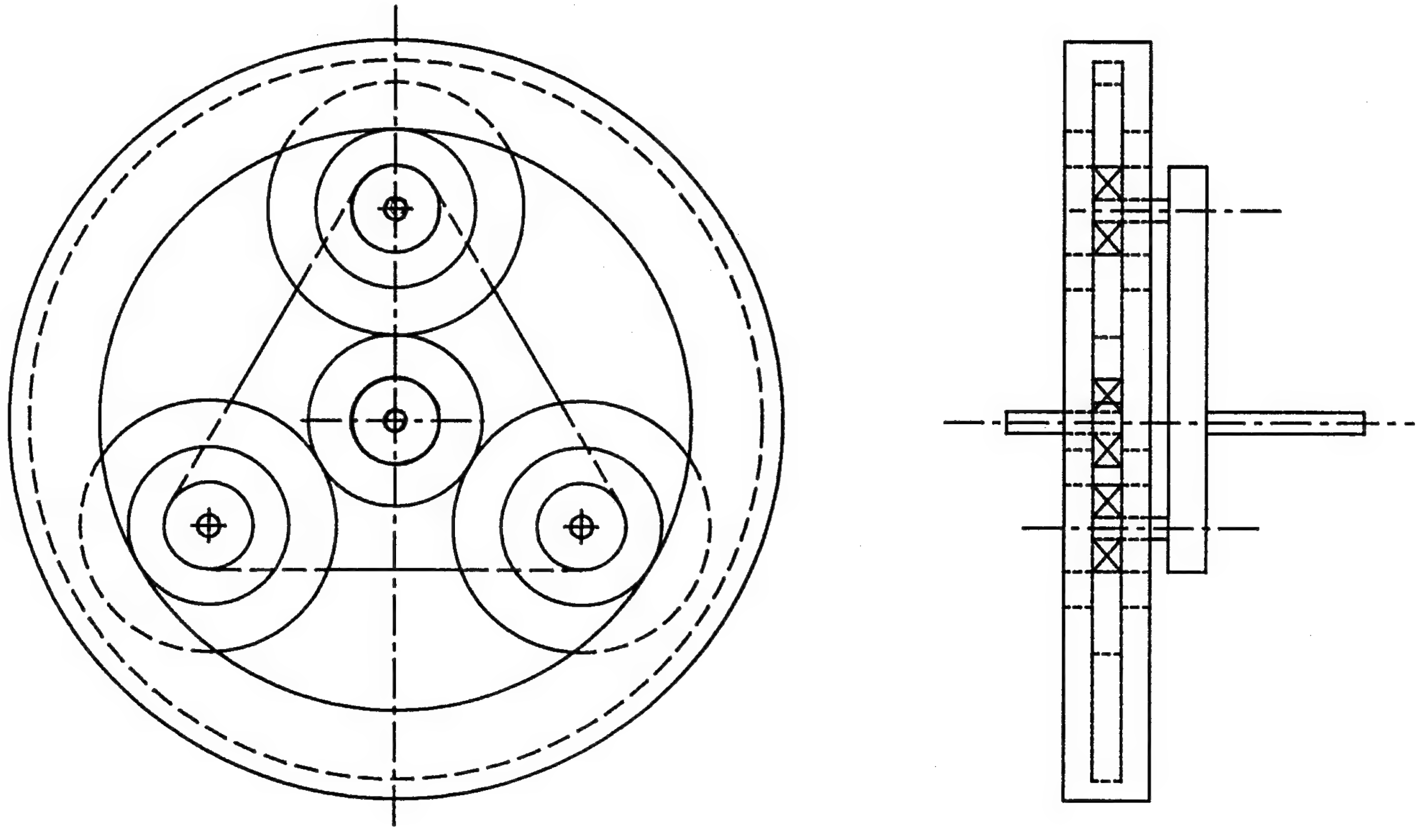


FIGURE 29 — SINGLE–PLANE REDUCTION WITH STEPPED PLANETS

given in equations (15) through (17) and (22) through (28) and the cycle count analysis is as given in Table 15 with N_p being the number of planets.

Spiral-Bevel Reductions

The spiral-bevel reductions of cases 10 and 11 complete the unit transmissions. Figure 11 shows some of the geometry which is required for the analysis of this gear mesh. The size of the bevel gear mesh is the cone distance, A_o, which is measured from the apex of the cones to the back side of the tooth face along the pitch line of the two mating gears. The face width of the gear mesh, f, is also measured along the pitch line.

The geometry of the gears is defined by the following inputs: the shaft angle, Σ, between the input and output shaft center-lines, the spiral angle, ψ, of the mesh at the midpoint of the face width, the normal pressure angle, ϕ_n, the numbers of pinion and gear teeth, the pinion and gear addenda, the direction of pinion rotation and the spiral hand of the pinion. The direction of pinion rotation is taken as seen from the back side of the pinion looking towards the apex. The output gear can be an internal gear or an external gear, as shown in Figures 11 and 13, and the output rotation, as seen from the back of the output gear looking toward the apex, can be in the same or the opposite direction as the input rotation.

For the program, the choice of output gear type is really a choice of output gear rotation direction. A nominal choice of an external output gear is a choice of opposite output gear rotation or a negative gear ratio, n; while a choice of an internal output gear is a choice of output gear rotation in the same direction or a positive gear ratio, n. However, certain combinations of reduction ratio and shaft angle can cause an output gear rotating in the opposite direction to the input gear to be an internal gear.

At the transition point, the output gear is a crown gear with a cone angle of ninety-degrees. An external gear has a cone angle of less than ninety-degrees and an internal gear has a cone angle of more than ninety-degrees. An output gear which is an internal gear with the direction of rotation of the pinion and the output gears being opposite, with one turning clockwise about its shaft looking toward the apex and the other turning counter-clockwise about its shaft, is shown in Figure 15.

The cone angles of the gear and pinion are found as:

$$\tan \Gamma_g = \frac{\sin \Sigma}{\cos \Sigma - 1/n} \tag{29}$$

and,

$$\tan \Gamma_p = \left| \frac{\sin \Sigma}{\cos \Sigma - n} \right| \tag{30}$$

The pitch point of the mesh is located at the midpoint of the gear face along the pitch line. The distance of the midpoint of the gear face from the apex of the mesh is:

$$D_o = A_o - f/2 \tag{31}$$

The pitch diameters of the gear and pinion at the pitch point for the spiral-bevel gears may be determined by:

$$D_p = 2.0\, D_o \sin \Gamma_p \tag{32}$$

and,

$$D_g = 2.0\, D_o \sin \Gamma_g \tag{33}$$

Figure 30 shows the spiral-bevel gear in the pitch plane. The pitch plane is defined as the surface that is tangent to the pitch cones of the meshing gears at the pitch point and perpendicular to the plane containing the axes of the gears.

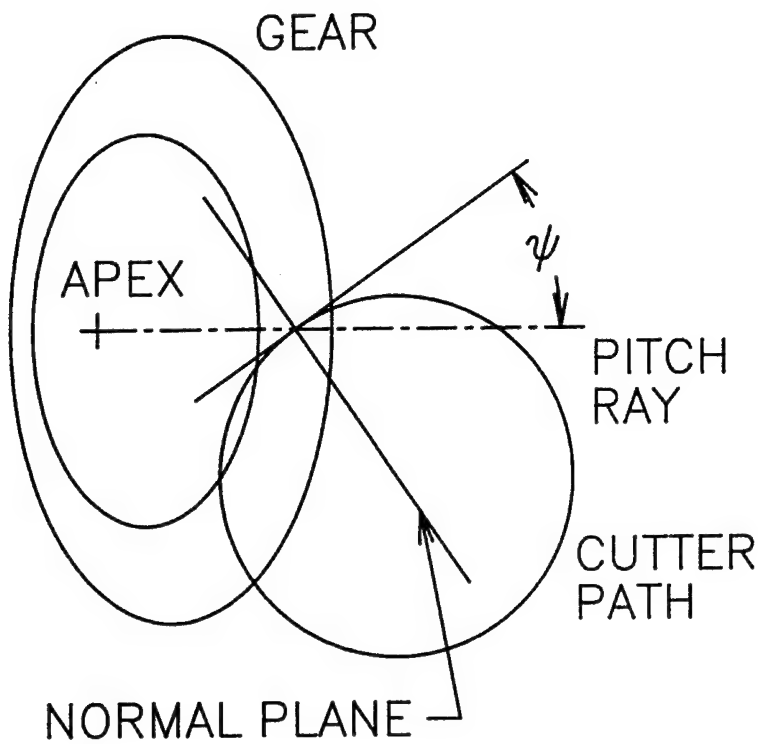


FIGURE 30 — PITCH PLANE OF A SPIRAL-
BEVEL GEAR

The spiral angle, ψ, of the mesh at the midpoint of the face width of the gear mesh is as shown. The angle is positive for a right-hand spiral and negative for a left-hand spiral. The hand of the spiral is defined by the direction of advance of the teeth from the base of the tooth towards the apex. Figure 30 shows a left-hand spiral. In the spiral-bevel gear mesh, the spiral angle of the gear and pinion must be the same, but the hands of the meshing gears are opposite for an external output gear and the same for an internal output gear.

To treat the strength of the teeth in mesh, the gears are modeled as equivalent planar spur gears in the normal plane. The diameter of the equivalent spur gear for a straight bevel gear is twice the back cone distance of the gear. The back cone distance of the gear is the perpendicular distance from the pitch-point of the gear to the gear axis. For the case of spiral-bevel gears, the back cone distance includes the effects of the spiral angle. The equivalent spur-gear diameters are found including the spiral-angle effect as:

$$D_{pe} = \frac{D_p}{2.0 \cos \Gamma_p \cos^2 \psi} \tag{34}$$

and

$$D_{ge} = \frac{D_g}{2.0 \cos \Gamma_g \cos^2 \psi} \tag{35}$$

In accordance with the AGMA design standard [17], the face width of the equivalent spur gear is given by:

$$f_e = \frac{f\, Z\, \eta_I \cos \psi_b}{\eta^2\, m_N} \tag{36}$$

where η and η_I are distances along a vectorized line of action and m_N is the gear mesh load sharing ratio. The length of the line of action, Z, is calculated for the equivalent spur gears and the radii of curvature of the

contacting teeth are taken at the point of highest contact stress on the equivalent spur gears, as found by numerical iteration. These equations can also be used for the case of straight bevel gears.

The module or diametral pitch of the teeth is modeled at the mid-tooth pitch radius for planar spur gears and may be found in terms of the number of teeth and the pitch cone geometry as:

$$m = \frac{2.0\ D_o \sin \Gamma_g}{N_g} \tag{37}$$

or,

$$P_d = \frac{N_g}{2.0\ D_o \sin \Gamma_g} \tag{38}$$

depending on the system of units.

The direction of rotation of the pinion is required to determine the loads transmitted at the contact point. The analysis assumes that the pinion is the driver, and that the direction of rotation of the gear is defined by looking at the gear from the base of the gear towards the apex of the mesh.

The pinion and the bearings supporting it rotate at a speed different from the output shaft, so they see a different number of load cycles from that of the output shaft. The load cycles for the pinion and the bearings supporting it can be found by standard motion analysis as R_o/R_i times the number of output shaft rotations.

Dual Spiral-Bevel Reductions

For the dual spiral-bevel reduction of case 11, both input pinions are assumed to be identical in all respects. The location of the two inputs is specified by the angle Λ, between the input pinions in the plane of the axes of the two pinions as shown in Figure 16. The geometry of the mesh is identical to that of the spiral-bevel reduction, using equations (29) through

(38). The geometry of the dual spiral-bevel reduction with an internal output gear is shown in Figure 15 with the pinions and output gear rotating in opposite directions.

The output gear sees two load cycles due to the two input pinions, while its support bearings see a single cycle of a combined load. The pinions and input bearings see one load cycle for their own shaft rotation. The ratio of the number of load cycles of the input pinion to the output gear is again given as R_o/R_i. This is also the reduction ratio for the transmission.

ASSEMBLY ANALYSES

The program makes three assembly checks for parallel reductions, reverted reductions, single-plane reductions and dual spiral-bevel reductions. The first test is for concentricity of the input and output shafts for the reverted reductions and single-plane reductions. The second test is for circumference clearance for the planets and/or the intermediate gears for parallel reductions, reverted reductions and single-plane reductions. The last check is for adequate clearance between the two input pinions of the dual spiral-bevel reductions. If any of these tests uncovers a problem, a message is written to the output file with assembly information from the test and the program execution is terminated.

The concentricity test checks for equal center distance between the intermediate shafts and the input or output shaft center-lines. For proper assembly, the distances should not differ by more than a small quantity. The relation can be written as

$$(R_i + R_{ii}) - | R_o + R_{io} | \leq e \tag{39}$$

For the purpose of calculation, the pitch radius and tooth surface curvature of internal gears are taken as negative to indicate concavity. In equation (39), R_i and R_o are the input and output gear pitch radii, R_{ii} and R_{io} are the pitch radii of the intermediate gears which mesh with the input and output gears respectively and e is a small radial clearance equal to one percent of the input gear addendum. Subroutine TESTDI performs this check for reverted and single-plane reductions.

There must be adequate circumferential clearance between adjacent planets to prevent interference among their teeth. Figure 31 shows two adjacent planets in contact with an input gear. The central angle, Σ_i, is

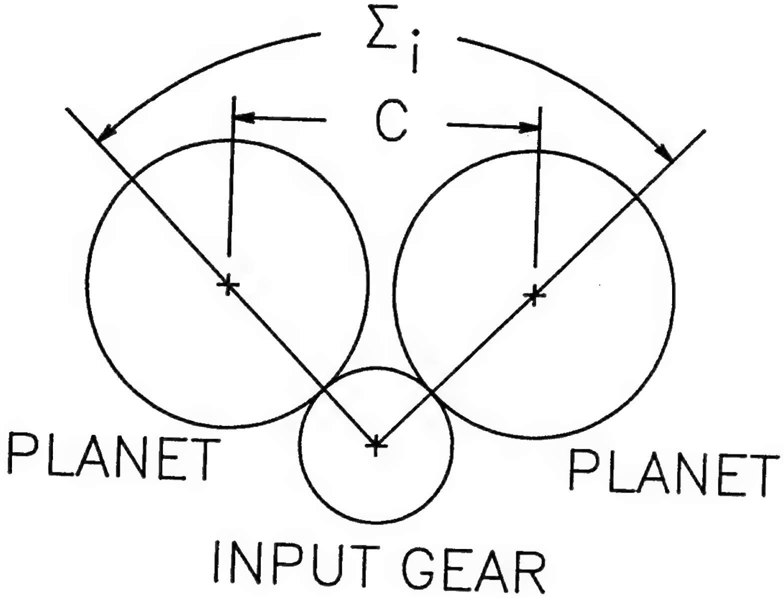


FIGURE 31 – CIRCUMFERENTIAL CLEARANCE GEOMETRY

between two adjacent planet center-lines with the input shaft center. A similar test can be made with the output gear's planets. From the geometry of Figure 31, the distance C, between the axes of two adjacent planets is given as:

$$C = 2.0 \cdot | R_i + R_{ii} | \sin (\Sigma_i/2) \tag{40}$$

where R_i is the pitch radius of the input pinion and R_{ii}, the pitch radius of the planet or intermediate gear. For adequate clearance, the distance, C, must be greater than the outside diameter of the planet or intermediate gear. Subroutine TESTCL checks this clearance for parallel, reverted and single-plane reductions.

In the dual spiral-bevel reduction, there must be adequate clearance between the two input pinions for proper meshing. Figure 32 shows the outer diameter of two input pinions and the output gear in the plane of the axes of the input pinions. The geometry required for calculating the clearance is shown in Figure 33. From the geometry of the Figure, we have the addendum angle of the pinion, $\delta\theta$, in radians as:

$$\delta\theta = \frac{a}{A_0} \tag{41}$$

where a is the back cone addendum of the input pinion and A_0 is the back cone distance of the mesh. The effective cone angle for the outside diameter of the input pinions in radians is then:

$$\Gamma_{eff} = \Gamma_p + \delta\theta \tag{42}$$

And for adequate clearance, the angle between the input pinions must be greater than twice this effective cone angle of the pinion. Subroutine TESTDB performs this check for dual spiral-bevel reduction using this algorithm.

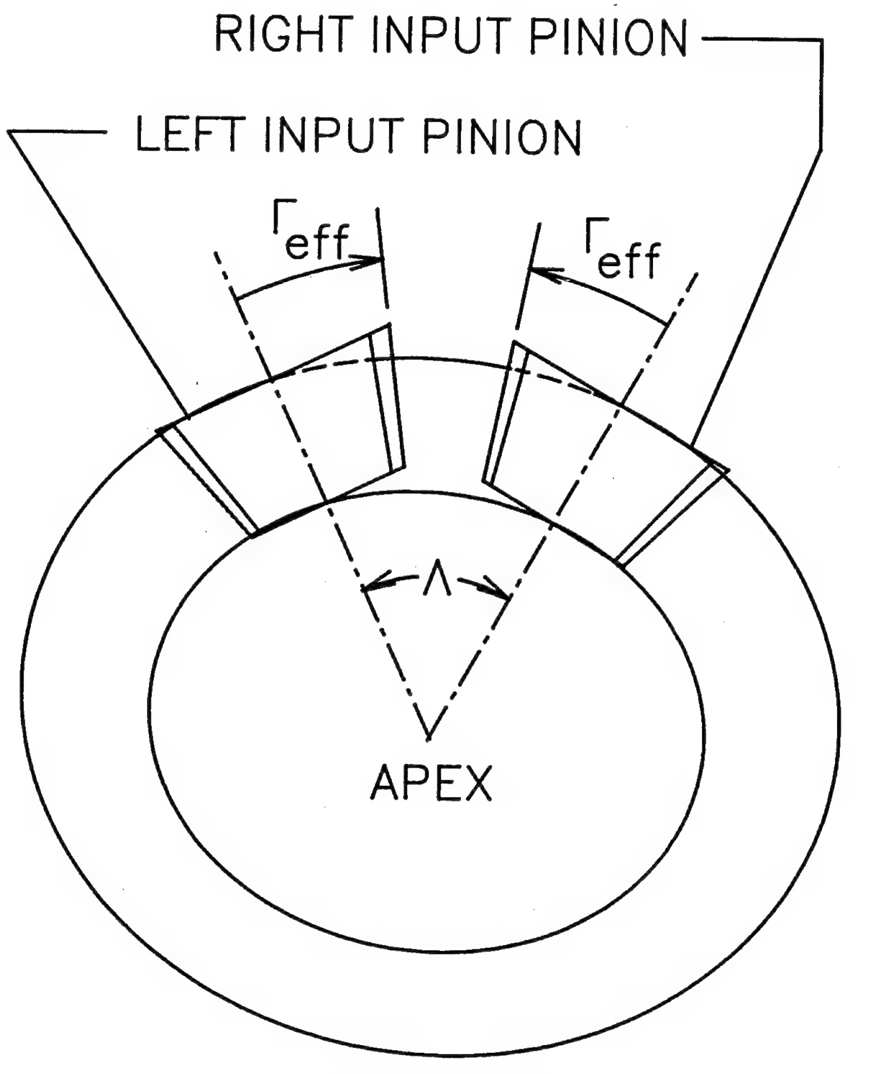


FIGURE 32 – SPIRAL–BEVEL INPUT–PINION CLEARANCE

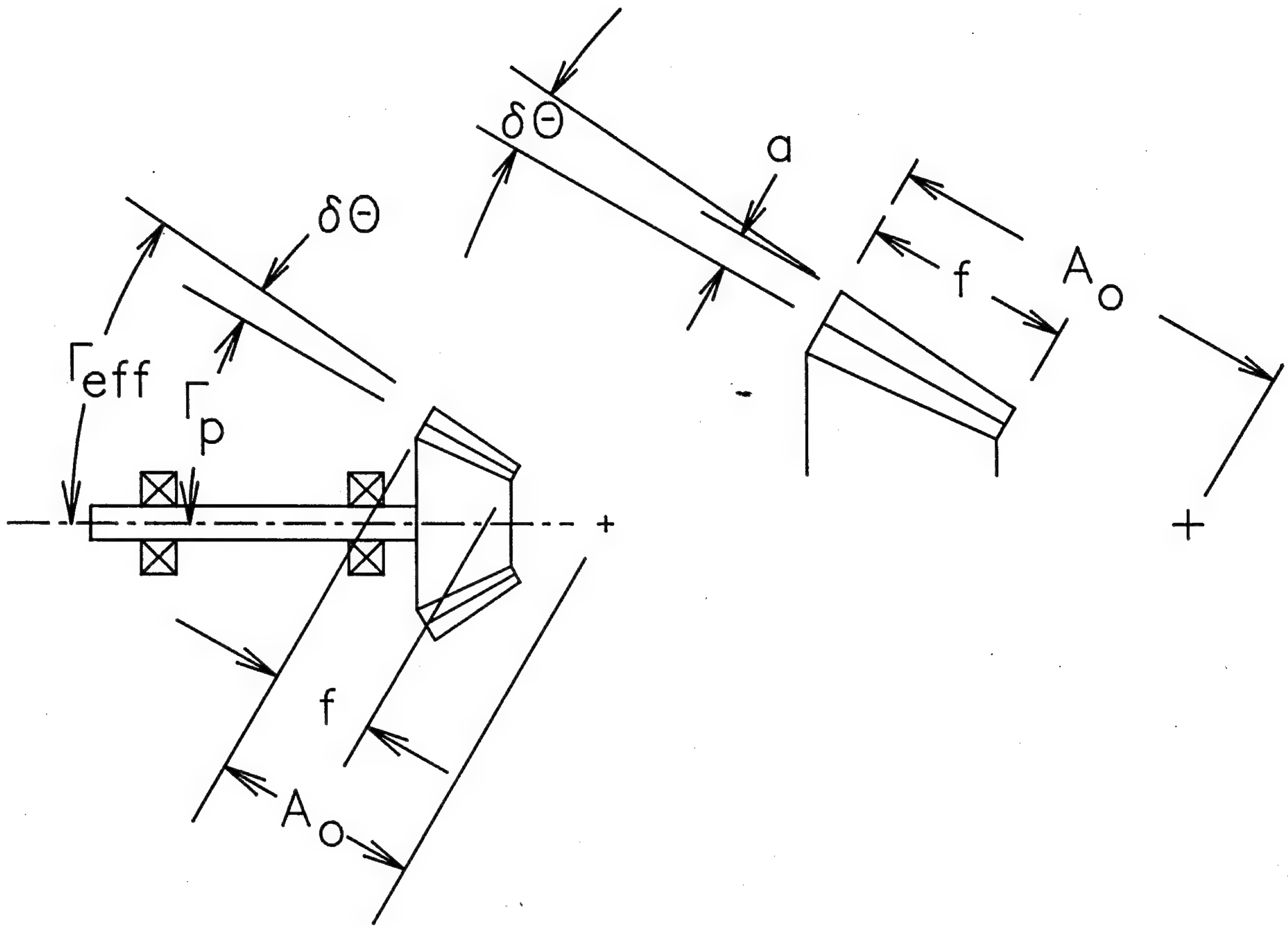


FIGURE 33 — INPUT—PINION ADDENDUM ANGLE

COMPONENT LOADING

The load on each of the component gears and bearings can be calculated from the geometry of the transmission and the input torque. The load calculations assume an efficient and well-lubricated gear train and equal load sharing among different parallel load paths. For all unit transmissions except the dual spiral bevel, the output torque is the overall speed ratio of the unit transmission times the input torque for the transmission.

$$T_o = n \cdot T_i \tag{43}$$

For the dual spiral-bevel unit transmission, the output torque is twice this value. The forces on the intermediate gears are equal to the forces on the input gear and the final gear depending on whether the intermediate gear is on the input side or output side respectively.

<u>Gear Mesh Forces</u>

For a spur-gear mesh, there are two gear force components: the tangential force and the radial force. The tangential force, F_t, can be calculated as the torque divided by the gear radius, R, and the number of parallel load paths, N_p.

$$F_t = \frac{T}{N_p \cdot R} \tag{44}$$

And the gear radial force, F_r, is calculated as the tangential gear force times the tangent of the normal pressure angle, ϕ_n.

$$F_r = F_t \tan \phi_n \tag{45}$$

The total gear mesh force, F_n, is given by the vector resultant of the two orthogonal radial and tangential force components as:

$$F_n = [\, F_t^2 + F_r^2 \,]^{1/2} \tag{46}$$

For a helical-gear mesh, the helix angle adds a third, axial component to the gear force. The tangential component is given by equation (44) and the radial and axial components are:

$$F_r = \frac{F_t \tan \phi_n}{\cos \psi} \tag{47}$$

and

$$F_a = \pm F_t \tan \psi \tag{48}$$

where the axial component is positive for an input pinion with a right-hand helix which is turning counter-clockwise as seen from the input side. It is also positive for an input pinion with a left-hand helix which is turning clockwise. Reversing just the hand or the rotation direction reverses the sign in equation (48). Figure 34 shows the mesh forces acting on a pinion with a right-hand helix which is turning counter-clockwise. The forces on the driven gear have the same magnitude and sign as those on the mating pinion. Using this sign convention, a positive axial gear force on the input and output shafts is directed out of the unit transmission towards the first support bearing as shown in Figure 34. The total gear force is the vector sum of the three forces:

$$F_n = [\, F_t^2 + F_r^2 + F_a^2 \,]^{1/2} \tag{49}$$

For a spiral-bevel gear mesh, there also are three force components, F_t, F_r and F_a, acting on the gear face, as shown in Figure 35 in their positive directions. As with the helical-gear mesh forces, these forces are dependent on the direction of the spiral hand, the direction of rotation of the gear and whether the gear is a driver gear or being driven. The gear forces are calculated using the formulas given by Dudley [18], as:

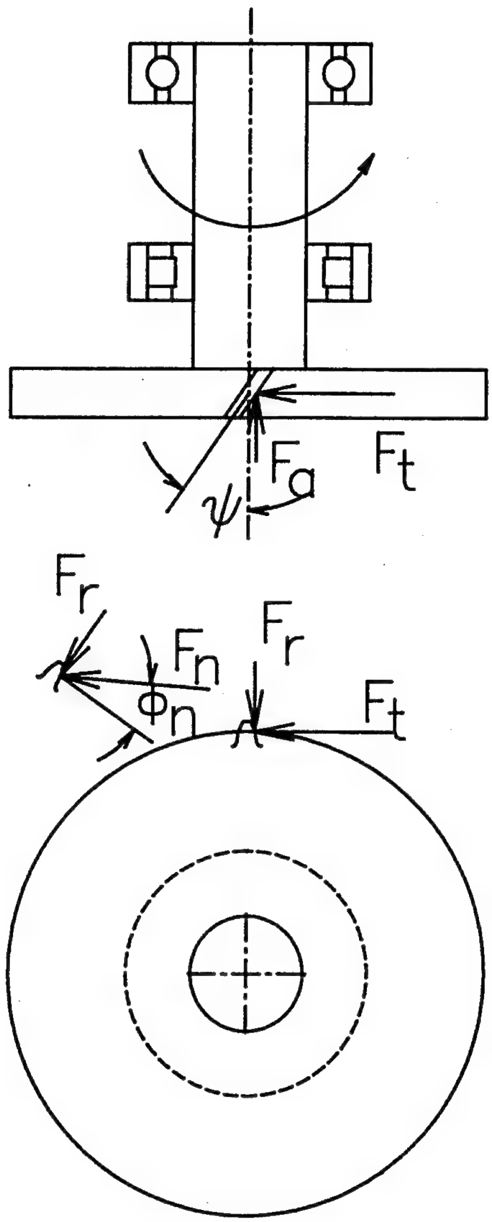


FIGURE 34 — HELICAL—PINION GEAR—MESH FORCES

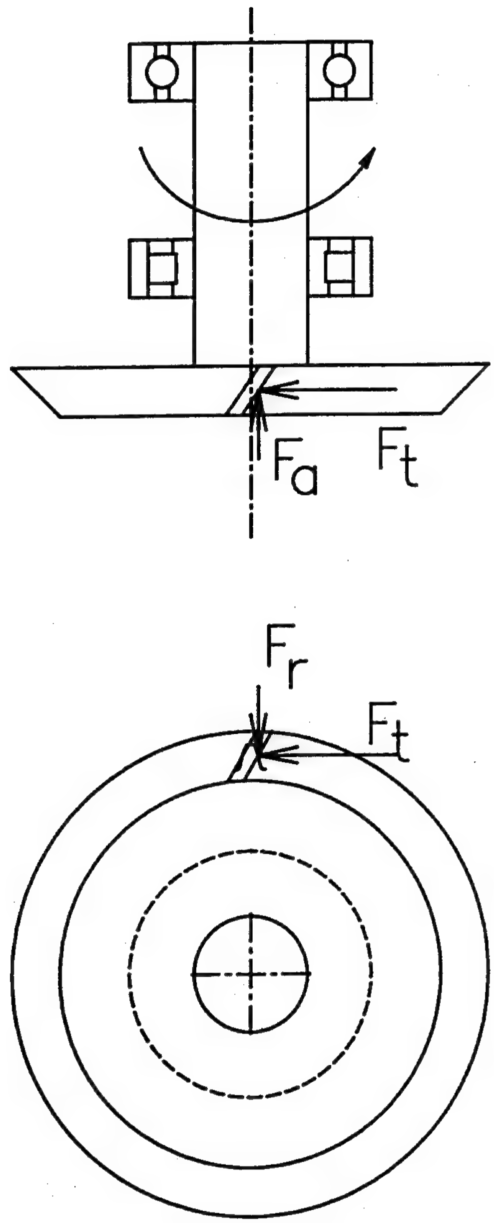


FIGURE 35 — SPIRAL—BEVEL
GEAR—MESH FORCES

$$F_t = \frac{T}{D_o \sin \Gamma_g} \tag{50}$$

$$F_r = F_t \frac{\tan \phi \cos \Gamma_g \pm \sin \psi \sin \Gamma_g}{\cos \psi} \tag{51}$$

and,

$$F_a = F_t \frac{\tan \phi \sin \Gamma_g \mp \sin \psi \cos \Gamma_g}{\cos \psi} \tag{52}$$

Equations (51) and (52) are for a driving gear with a right-hand spiral which is rotating clockwise or a left-hand spiral which is rotating counter-clockwise. For a driven gear, the equations are for a right-hand spiral which is rotating counter-clockwise or a left-hand spiral which is rotating clockwise. For the four other cases the sign of the last term is switched. It may be noted that for a shaft angle of 90 degrees, the radial load of the pinion will be equal to the axial load of the gear, also the axial load of the pinion will be equal to the radial load of the gear. The total resultant load, which is given by equation (49), and the tangential load on the gear and pinion will be the same for all shaft angles.

Subroutine LOAD calculates these gear tooth forces.

Single-Gear Support Bearing Loading

For a shaft which supports a single gear, there are two possible bearing mounting configurations: straddle and overhung. The tangential and radial bearing loads on an input or output shaft are found from the tangential, radial and axial gear loads and the bearing mounting distances A and B. The distance A is taken as positive for a straddle mounting and negative for an overhung mounting. The gear and bearing loads are shown in Figure 36 for the

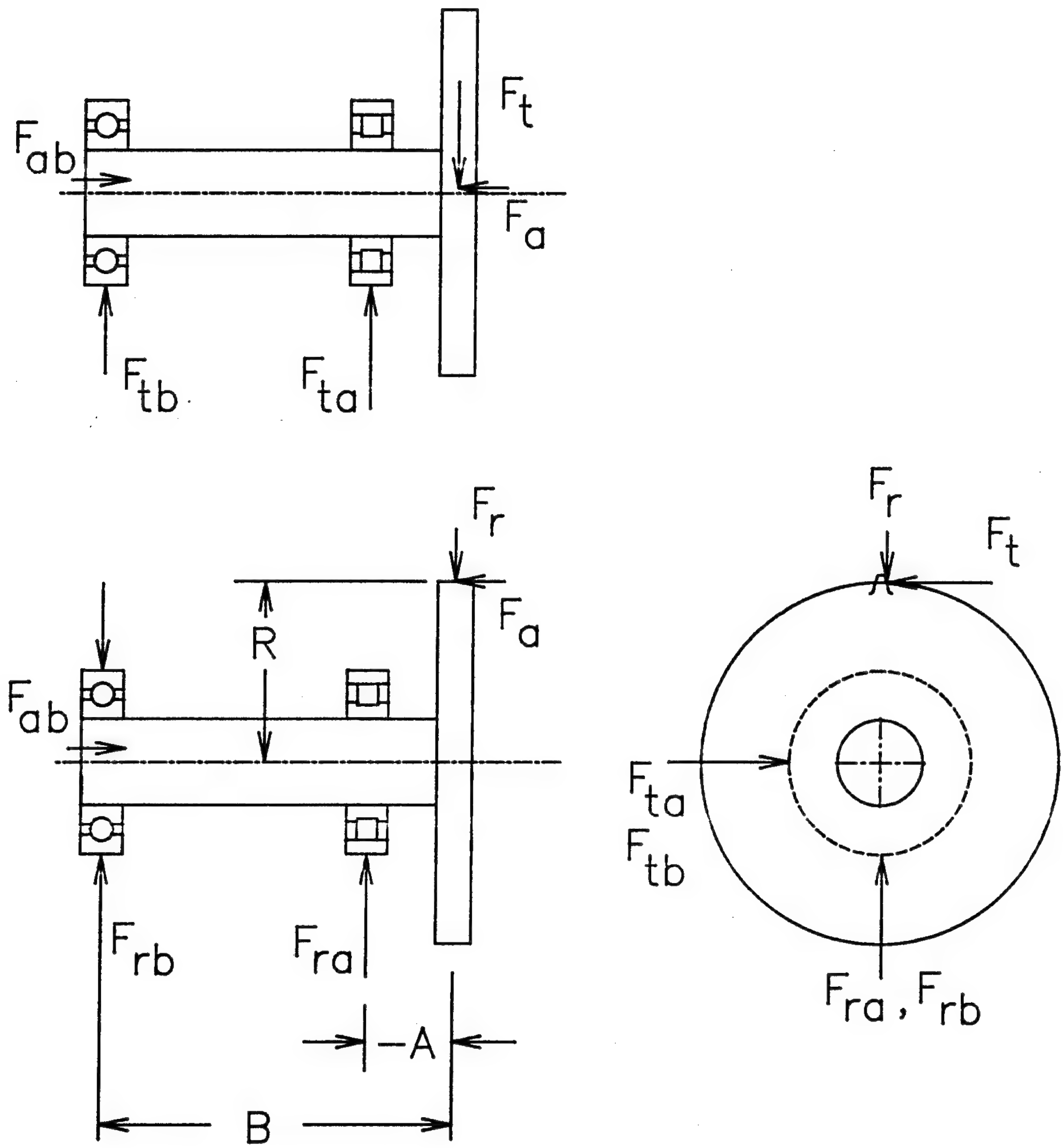


FIGURE 36 – SINGLE-GEAR OVERHUNG SUPPORT LOADING

overhung configuration, and in Figure 37 for the straddle configuration. The following analysis is valid for both configurations.

The axial load on the bearing that can support thrust loads is equal to the axial gear load unless the two bearings share this load, in which case it is assumed that each bearing carries one-half of the gear axial load.

The tangential bearing loads are taken as the components of the radial loads on the bearings which act in the plane of the tangential load on the gear mesh. These tangential bearing reactions are:

$$F_{ta} = F_t \frac{B}{A + B} \tag{53}$$

and

$$F_{tb} = F_t \frac{A}{A + B} \tag{54}$$

These reactions act in the same plane in a direction opposite to the gear force component.

The radial bearing loads are the components of the radial loads on the bearings which act in the plane of the radial gear load. These radial bearing reactions, which also support moments from the axial gear load, are:

$$F_{ra} = \frac{F_r B + F_a R A/|A|}{A + B} \tag{55}$$

and

$$F_{rb} = \frac{F_r A - F_a R A/|A|}{A + B} \tag{56}$$

As with the tangential bearing forces, these radial forces are positive when they oppose the radial gear force.

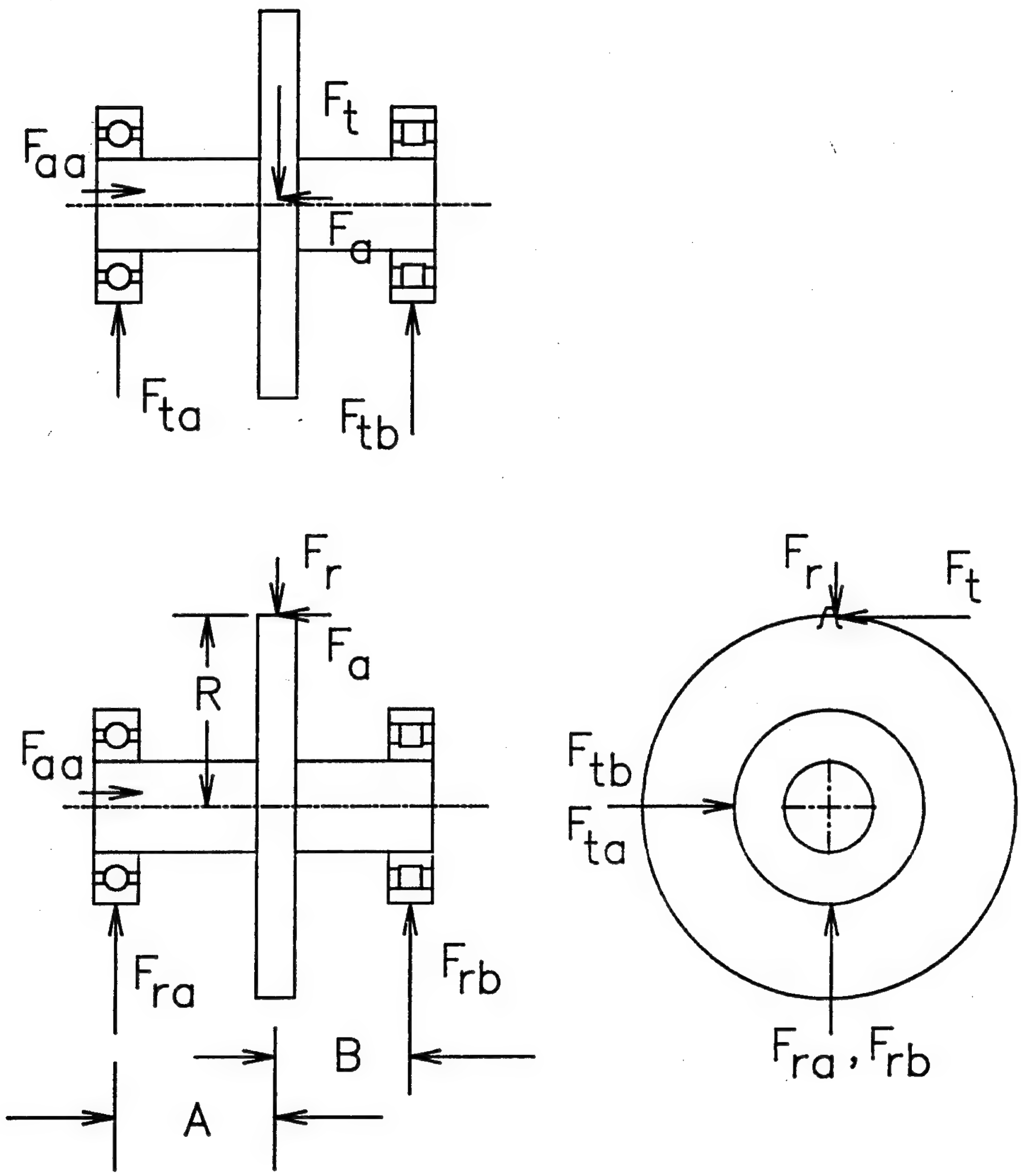


FIGURE 37 – SINGLE–GEAR STRADDLE SUPPORT LOADING

The resultant radial bearing reactions are the vector sums of the radial and tangential components at each bearing. For the bearing a distance A from the gear, this resultant is:

$$F_{total,a} = [\, F_{ta}^{\;2} + F_{ra}^{\;2} \,]^{1/2} \tag{57}$$

When the single gear on this shaft is in mesh with two gears which are separated by a shaft angle; Σ_i, Σ_o or Λ, the tangential direction on the gear is taken midway between the two contacting gears and the gear forces and pitch radius are changed in the equations:

$$F_{te} = 2.0\, F_t \cos(\, \Sigma_i/2 \,) \tag{58}$$

$$F_{re} = 2.0\, F_r \cos(\, \Sigma_i/2 \,) \tag{59}$$

$$F_{ae} = 2.0\, F_a \tag{60}$$

and,

$$R_e = R \cos(\, \Sigma_i/2 \,) \tag{61}$$

Figure 38 shows this geometry for an input gear and subroutine BLC1 calculates these bearing reactions.

The input and output gears for the reverted reductions and for the single-plane reduction carry no radial loads due to the assumed symmetry of the planet locations. For the helical reverted reductions, the input and output thrust bearings carry the sums of the axial loads from all the sun-planet and output gear-planet meshes.

Intermediate Shaft Bearing Forces

The intermediate shafts support two gears with two bearings as shown in Figure 39. To show the reactions on the intermediate shaft bearings, Figure 39 gives three views of the intermediate shaft with a double straddle mounting, including an end view as seen from the input side. By making the distances from the bearings to the gears negative for bearings which lie between the two gears, this analysis is also valid for the other three

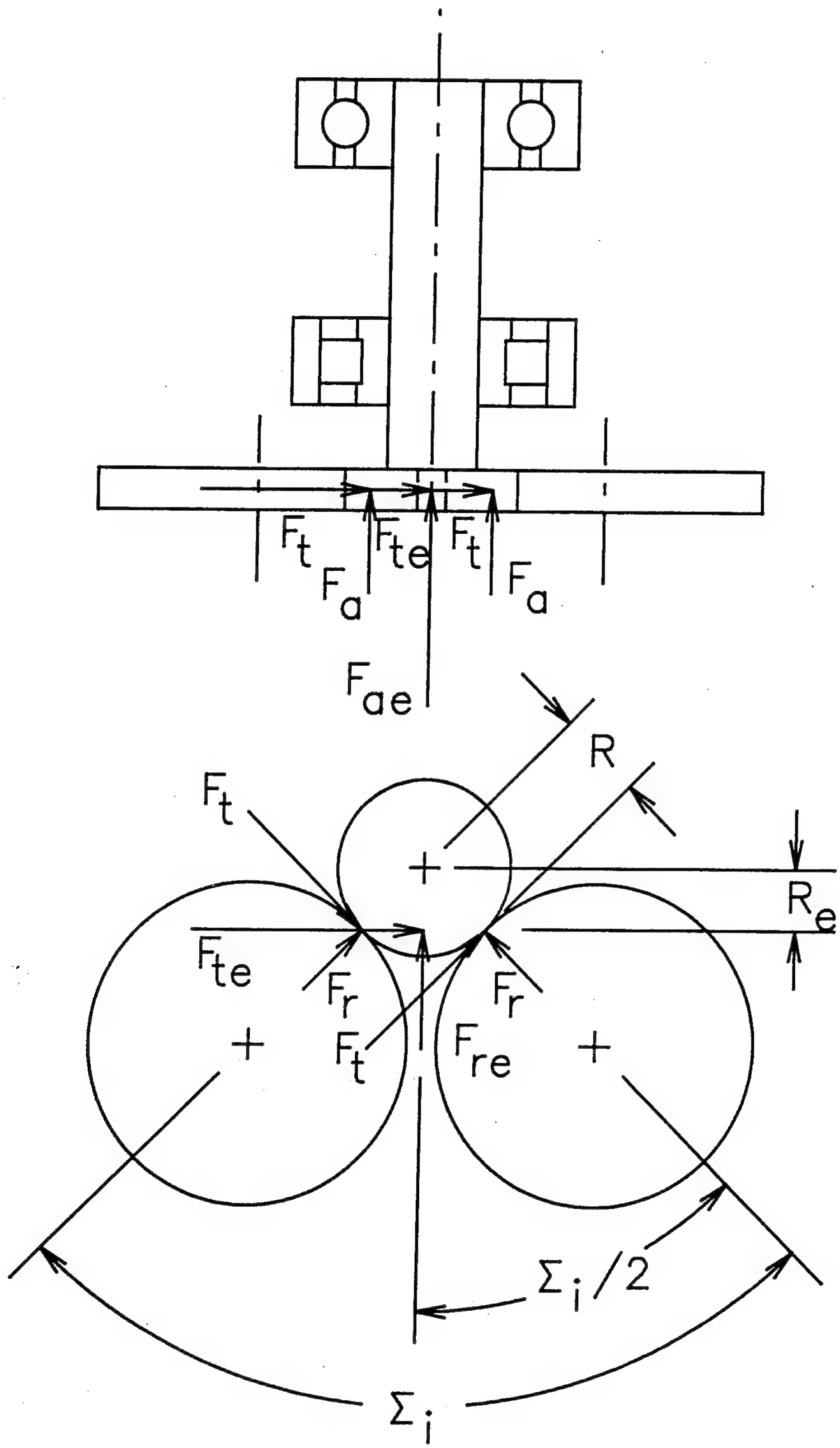


FIGURE 38 — TWO—GEAR LOAD SUMMATION INTO EQUIVALENT FORCES AND RADIUS

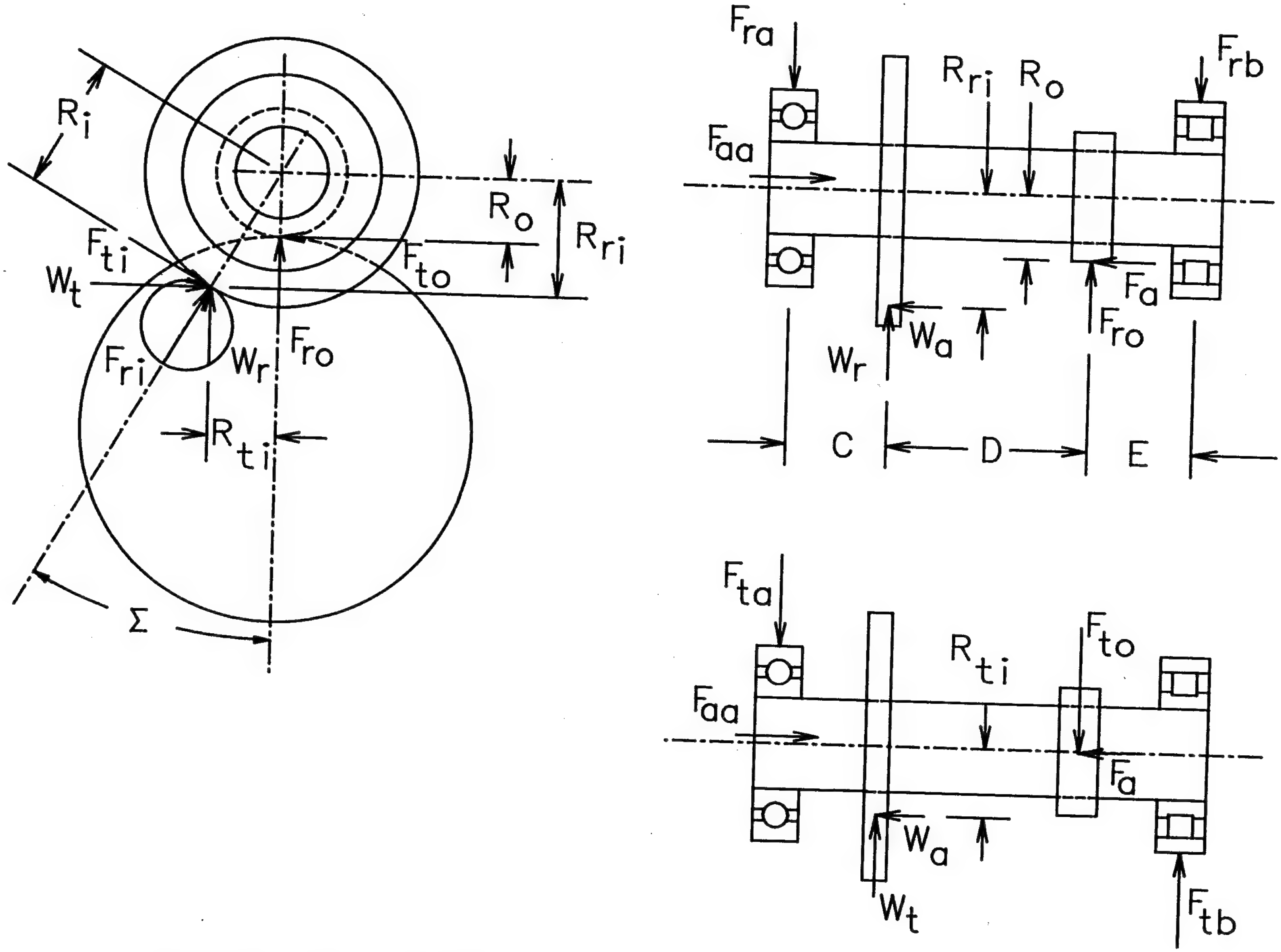


FIGURE 39 — INTERMEDIATE—GEAR SUPPORT LOADING

mountings. A positive shaft angle, Σ, is measured counter-clockwise from the input shaft location to the output shaft location about the intermediate shaft as seen from the input side.

For helical transmissions, the axial forces on both gear meshes are taken as positive for thrust loads directed toward the input. The axial reaction on the thrust bearing is the sum of the two axial gear forces. If the two bearings share this thrust load, it is assumed that they share it equally so that each bearing will see one-half the sum of the two axial gear forces.

As with the single-gear support analysis, the analysis for the radial bearing loads is performed in the directions of the tangential and radial forces on the output gear mesh. Resolving the input gear reactions and radial distances into forces and distances in these directions gives:

$$W_t = F_{ri} \sin \Sigma + F_{ti} \cos \Sigma \tag{62}$$

$$W_r = F_{ri} \cos \Sigma - F_{ti} \sin \Sigma \tag{63}$$

$$R_{ti} = R_i \sin \Sigma \tag{64}$$

and

$$R_{ri} = R_i \cos \Sigma \tag{65}$$

Summing moments about the first bearing gives equations for the loads on the second bearing:

$$F_{tb} = \frac{- C\ W_t + (C + D)\ ST\ F_{to} + R_{ti}\ W_a}{C + D + E} \tag{66}$$

and,

$$F_{rb} = \frac{C\ W_r + (C + D)\ ST\ F_{ro} - R_{ri}\ W_a - R_o\ ST\ F_a}{C + D + E} \tag{67}$$

where ST is a sign term which is plus one for an external final gear and minus one for an internal ring final gear. When the second gear meshes with an

internal gear, the output loads act on the opposite side of the second gear in the opposite direction.

The tangential and radial loads on the first bearing are now found by force equilibrium to be:

$$F_{ta} = W_t - ST\ F_{to} + F_{tb} \tag{68}$$

and

$$F_{ra} = W_r + ST\ F_{ro} - F_{rb} \tag{69}$$

In these equations, the components of the radial bearing forces are positive when they oppose their adjacent gear force. Although Figure 39 shows this force analysis for counter-clockwise rotation of the intermediate shaft, it is also valid for clockwise rotation. The resulting radial bearing loads are found as a vector sum of these two orthogonal components using equation (57).

Subroutine BLC2 calculates the intermediate shaft bearing reactions using these formulae.

Planet Bearing Forces

All the planets in a single-plane transmission are assumed to share the load equally. The tangential force on each sun-planet gear mesh is calculated for a spur-gear mesh from equation (44). The tangential force on the ring gear or the planet-ring gear mesh is found as:

$$F_{tr} = \frac{R_{ps}}{R_{pr}} \cdot \frac{T_i}{N_p\ R_s} \tag{70}$$

Figure 40 shows the loads on the planet gear from the sun-planet mesh and the planet-ring meshes. The total tangential force on the bearing is the resultant of the two tangential gear forces. As the forces act in the same direction, the resultant is given by:

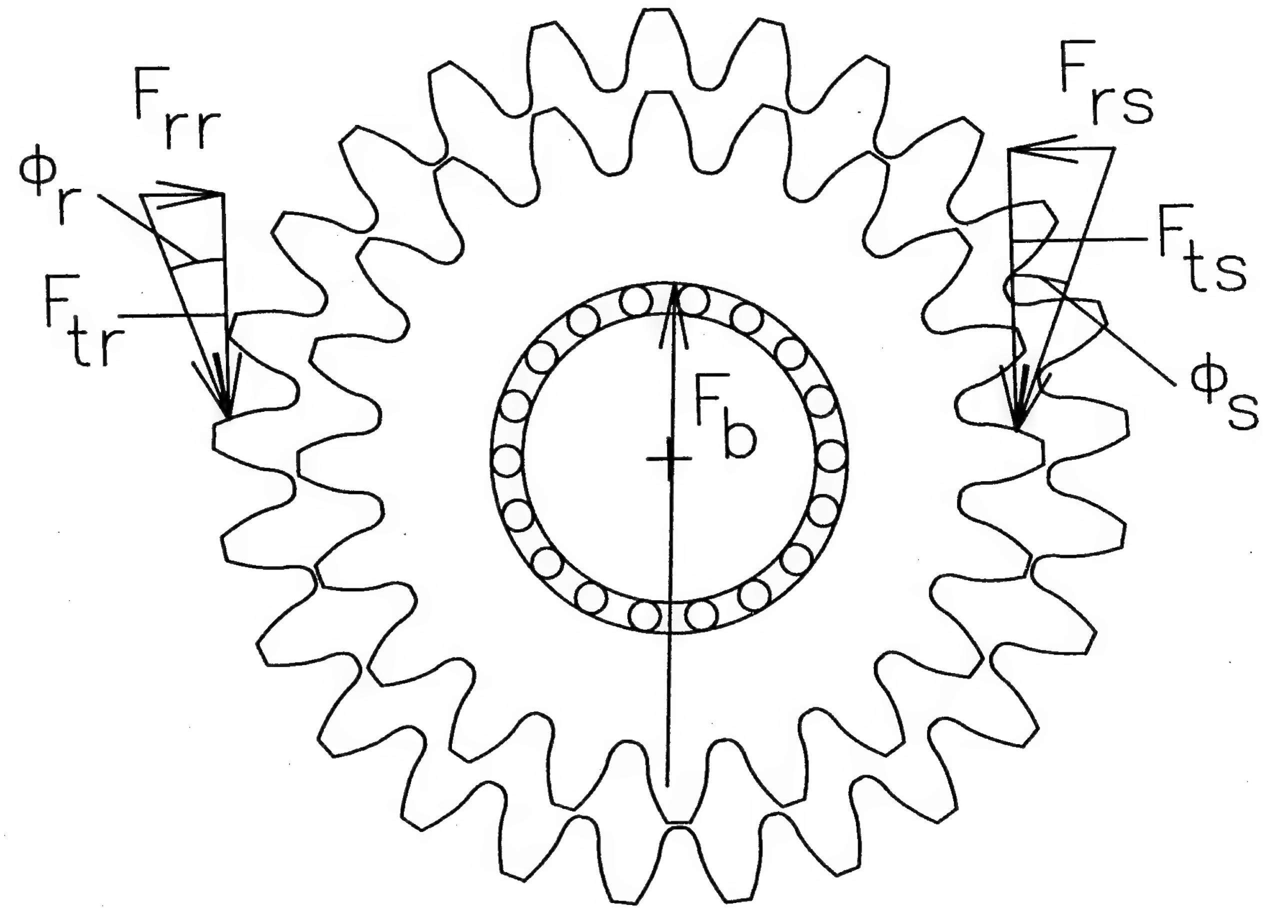


FIGURE 40 — PLANET FORCES

$$F_t = F_{tr} + F_{ts} \tag{71}$$

The radial forces in the two gear meshes may be found using the normal pressure angle ϕ, from equation (45). The total radial force on the planet gear bearing is the difference of these two radial forces for the sun-planet mesh and the planet-ring gear mesh, which is:

$$F_r = F_{tr} \cdot \tan \phi_{nr} - F_{ts} \cdot \tan \phi_{ns} \tag{72}$$

The total bearing reaction for the planet gear bearing is given by the vector sum of the radial and tangential force components. This is found using equation (46), where the radial and tangential forces are the resultant forces acting on the planet gear bearing.

Unit Transmission Interface Bearing Forces

In combining two unit transmissions which have two support bearings on the output shaft of the first unit and two on the input shaft of the second unit, care is take to ensure that these bearings are treated as the same bearings. They are entered twice in the input data because the program analyzes each unit separately. However, the two sets must be identical for the program to properly calculate the system life. The second bearing on the output shaft of the first unit should be the same as the first bearing on the input shaft of the second unit. Likewise the first bearing on the output shaft of the first unit should be the same as the second bearing on the input shaft of the second unit and the distance between the bearings should be the same.

If these conditions are satisfied, the bearings can be treated as the same and their loads superimposed before the final life analysis is performed. In this final life analysis, the lives of the output bearings of the input unit are ignored and the lives of the bearings are taken as the lives of the

input bearings of the second unit with the loads from both gears included. In addition to making the bearings descriptions consistent, the user must enter an angle, β, between the radial load on the output gear of the first unit and the radial load on the input gear of the second unit. The angle is measured counter-clockwise from the first unit gear load to the second unit gear load about the input shaft of the second unit looking into the second unit. If one of the two units has two gear meshes on its gear, the resulting radial gear load acts midway between the two gear meshes and is the load to which the angle, β, is measured.

Figure 41 shows these gear forces on a shaft which is turning counter-clockwise. It also shows the directions of the resulting positive tangential and radial bearing reactions on the two shafts. The equations for the superimposed bearing loads on each bearing are:

$$F_a = F_{a2} - F_{a1} \tag{73}$$

$$F_t = F_{t2} \mp F_{r1} \sin \beta - F_{t1} \cos \beta \tag{74}$$

and,

$$F_r = F_{r2} + F_{r1} \cos \beta \mp F_{t1} \sin \beta \tag{75}$$

where the subscript 2 indicates bearing loads from the second unit transmission and the subscript 1 indicates bearing loads from the first unit transmission. The signs of the two sine terms are negative for counter-clockwise shaft rotation and positive for clockwise rotation. This load adjustment is done in the main program after the unit analyses have been completed.

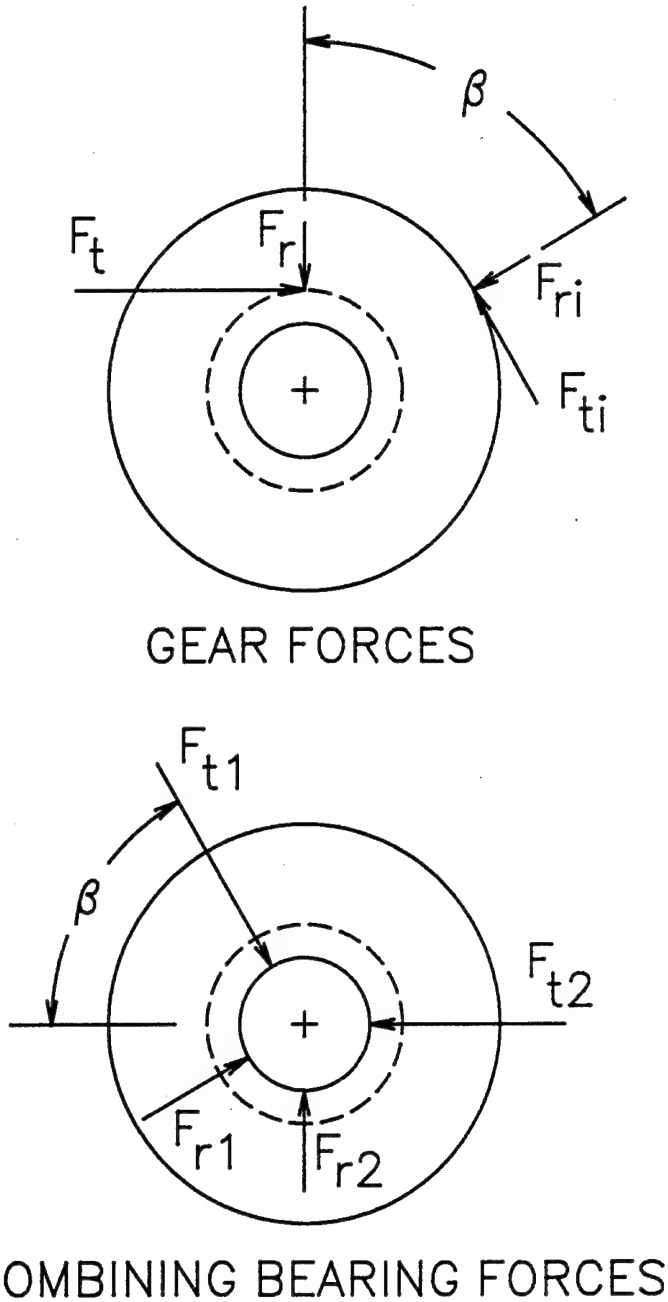


FIGURE 41 — BEARING FORCE SUPERPOSITION

LIVES AND DYNAMIC CAPACITIES

Surface pitting due to fatigue is the basis for the life model for the bearings, gears and unit transmissions. The life model was proposed by Lundberg and Palmgren in the 1930's for rolling element bearings [11,12]. They assumed that the natural logarithm of the reciprocal of reliability, R, of a bearing element is proportional to its life, ℓ, and stress parameters such as: the stress level, τ_0, maximum shear stress depth, z_0, and the stress volume, V. This relationship is:

$$\mathrm{Ln}\left(\frac{1}{R}\right) \propto \tau_0^{\,c}\, z_0^{\,-h}\, V\, \ell^{b} \tag{76}$$

where b is the Weibull slope and c and h are constants that are determined experimentally. This can be reduced to the standard two-parameter Weibull distribution for the scatter in the life:

$$\mathrm{Ln}\left(\frac{1}{R}\right) = \left(\frac{\ell}{\theta}\right)^{b} \tag{77}$$

where b is the Weibull slope and θ is the characteristic life of the distribution. In terms of the ninety-percent reliability life, ℓ_{10}, the two-parameter Weibull distribution can be represented as:

$$\mathrm{Ln}\left(\frac{1}{R}\right) = \mathrm{Ln}\left(\frac{1}{0.9}\right)\cdot\left(\frac{\ell}{\ell_{10}}\right)^{b} \tag{78}$$

This life to reliability relationship is for a specific load, F, at which the ninety-percent reliability life is ℓ_{10}. This load is related to the component dynamic capacity, C, as:

$$\ell_{10} = \left(\frac{C}{F} \right)^{p} \tag{79}$$

Here the component dynamic capacity, C, is defined as the load that produces a life of one-million cycles with a reliability of ninety-percent [11,12]. Since the dynamic capacity life is one-million cycles, it does not appear in the equation. The power, p, is the load-life exponent which is determined experimentally.

Bearing Life Model

The ASME rolling element committee of the ASME lubrication division [15] has modified equation (79) to cover many different situations with adjustment factors. The modified equation is:

$$\ell_{10,a} = a \left(\frac{C}{V\ F_r} \right)^{p} \tag{80}$$

where $\ell_{10,a}$ is the adjusted ninety-percent reliability life of the bearing, a is the life adjustment factor, V is the load adjustment factor, and F_r is the equivalent radial load that produces an equal amount of fatigue damage as the radial and thrust loads acting on the bearing. The life adjustment factor is a product of a number of factors which influence the life of bearings. These factors are for material quality, material processing, lubrication, speed and bearing misalignment. The factors for material quality, material processing and lubrication enable the designer to select bearings that are better than standard bearings. The factors for speed and misalignment are de-rating factors. These allow the designer to compensate for misalignments and speeds with which bearings are used.

The load adjustment factor is used to find an equivalent static load for comparing the dynamic capacity. This factor accounts for shock load and curvature correction. This is used to estimate the maximum load from the nominal design load.

The Weibull slope, b, is 10/9 for ball bearings, 9/8 for straight roller bearings and 3/2 for tapered roller bearings. The load-life exponent, p, may have different values such as 3.0 for ball bearings and 10/3 for roller bearings [15].

When axial loads are combined with radial loads in a ball bearing, the equivalent radial load is determined based on the ratio of the axial load to the basic static load rating, C_0, of the bearing with the relations given by Harris [16]. Routine BALL performs these calculations for single and double-row ball bearings using the routines LINT, DINT and SINT to interpolate the axial thrust effect tables.

For tapered-roller bearings, routine TAPER calculates the equivalent reaction for double-row tapered-roller bearings using the radial capacity of one of the bearing's rows and the radial to axial capacity ratio of the bearing.

Subroutine BDCAP performs the life and dynamic capacity calculations.

<u>Gear Life Model</u>

Gear tooth failures can be similar to those of the bearing elements [5,6]. There are a few differences due to the complex shape of the gear tooth. Most differences result in sudden tooth breakage in a lightly lubricated, overloaded mesh. This is not considered as a failure mechanism in this life model as the gears are assumed to be well-lubricated and not subjected to shock loading.

The American Gear Manufacturers Association (AGMA) publishes design codes for proper manufacture of gears [13,17]. These codes are aimed at reducing three modes of tooth failures: the gear tooth breakage caused by the bending loads on the tooth, scoring failures and surface pitting.

With proper selection of gear materials for the application and gear tooth sizing to withstand bending stresses it has been shown that failures from tooth breakage due to bending loads can be avoided. Proper lubrication can prevent gear tooth scoring.

The failure due to surface pitting of the gear tooth follows a pattern similar to rolling-element bearing race pitting. As the gear tooth life is similar to bearing life, engineers at the NASA Lewis Research Center have formulated a model for the gear tooth life based on the Lundberg-Palmgren model [5,6]. Starting from equation (76), Coy, Townsend and Zaretsky [5,6] developed a reliability model for the tooth pitting life of spur gears.

From equation (76), they obtained a relation for the dynamic capacity, C_t, of a spur-gear tooth. In terms of Buckingham's load stress factor, B [19,20], the dynamic capacity can be written as:

$$C_t = B \left[\frac{f_e}{\sum 1/\rho} \right] \tag{81}$$

Where the Buckingham's load stress factor, which has the dimension of stress, is a direct function of the material surface fatigue strength, S_{ac}:

$$B = 2 \pi S_{ac}^2 \left(\frac{1 - \nu^2}{E_\ell} \right) \tag{82}$$

This material surface fatigue strength, S_{ac}, includes the AGMA [13] surface strength, S_a, and several design factors:

$$S_{ac} = S_a \left(\frac{C_L \, C_H}{C_T \, C_R} \right) \left(\frac{C_V}{C_a \, C_s \, C_m \, C_f} \right)^{1/2} \tag{83}$$

The parameter f_e is the effective face width of the tooth and $\Sigma\, 1/\rho$ is the curvature sum for the pair of teeth in mesh at the point of contact. The curvature sum is:

$$\Sigma\left(\frac{1}{\rho}\right) = \left(\frac{1}{\rho_g}\right) + \left(\frac{1}{\rho_p}\right) \tag{84}$$

where, ρ_p and ρ_g are the radii of curvature of the pinion tooth surface and gear tooth surface at the contact point in the equivalent spur gear plane.

The dynamic capacity of the gear is lower than that of a single tooth. Each rotation of the gear subjects every tooth to a load cycle, and the gear fails when any gear tooth fails. As the accumulation of damage in any tooth is independent of the damage in any other tooth, the theory of series probability life can be applied. The reliability of the gear, R_g, can be expressed as the product of the reliabilities of each tooth in the gear. Thus:

$$R_g = (R_t)^{N_g} \tag{85}$$

where R_g is the reliability of the gear, R_t is the reliability of the tooth and N_g is the number of teeth on the gear. It is assumed that the reliability of any one tooth is equal to the reliability of any other tooth.

In terms of the lives of the tooth and the gear, substituting equation (85) into equation (78) for the two reliabilities gives:

$$\ell_{10,g} = \left(\frac{1}{N_g} \right)^{1/b} \ell_{10,t} \tag{86}$$

This relation is for gears in a single mesh only. For the case of collector gears such as with the parallel reduction, planetary gear trains and idler gears, the damage accumulates differently. As the load count will be different for these gears, the equation has to be modified to account for this variable loading. In planets and idler gears each tooth is loaded on both faces in one rotation. Since the fatigue damage accumulates separately on the faces, the gear faces are treated as separate gears in their own mesh in this simulation. The load count factor, ℓ_c, is used in other cases. This has units of load cycles per rotation. The more general form of equation (86) that accounts for different loadings on the tooth surfaces may be written as:

$$\ell_{10,g} = \frac{1}{\ell_c \, N_g^{1/b}} \cdot \ell_{10,t} \tag{87}$$

An equation for the dynamic capacity, C_g, of the gear can be found by substituting equation (79) into equation (87). This produces:

$$C_g = \frac{C_t}{\ell_c^{1/p} \, N_g^{1/(b \cdot p)}} \tag{88}$$

Helical and spiral-bevel gears are modeled as equivalent spur gears in the normal plane for these calculations. For helical gears, the pitch diameter of the equivalent spur gear is given by equation (3), its width is given by equation (9) or (10) and the radii of curvature at the contact point by equations (11) through (13). For spiral-bevel gears, the pitch diameters of the equivalent spur gears are given by equations (34) and (35), and the effective face width by equation (36).

For gears, the Weibull slope, b, for the examples of this report is taken as 2.5 based on the experimental data from NASA Lewis Research Center

[5,6]. The gearing load-life exponent, p, value of 8.93 in the examples is taken from the slope of the reliability factor versus life curve in the contact stress standard of the AGMA [13].

Subroutine SET performs the component life and dynamic capacity calculations using the function BSCAP for all spur gears.

Transmission System Life Model

For any system which needs service when any one of its components needs service, a strict series probability of all the component reliabilities can be assumed. The system reliability, R_s, is the product of the reliabilities of all the components, R_i.

$$R_s = \Pi R_i \tag{89}$$

The log of the reciprocals of equation (89) is:

$$\mathrm{Ln}\left(\frac{1}{R_s}\right) = \Sigma \, \mathrm{Ln}\left(\frac{1}{R_i}\right) \tag{90}$$

Substituting equation (78) into equation (90) for each component of the transmission yields :

$$\mathrm{Ln}\left(\frac{1}{R_s}\right) = \mathrm{Ln}\left(\frac{1}{0.9}\right) \Sigma \left(\frac{\ell_s}{\ell_{10,i}} \right)^{bi} \tag{91}$$

The life of the entire system is given by ℓ_s, and the reliability at this life is R_s. This is also the life of each component at the same system reliability, R_s. For consistency all the components must have the same counting base, which is chosen to be hours of operation in this simulation.

Equation (91) is not a true two parameter Weibull distribution as all the component Weibull slopes are not equal. However, a true two-parameter Weibull distribution can be used as a good approximation. The true

two-parameter Weibull distribution with the system reliability parameters can be written [20] as:

$$\mathrm{Ln}\left(\frac{1}{R_s}\right) = \mathrm{Ln}\left(\frac{1}{0.9}\right) \cdot \left(\frac{\ell_s}{\ell_{10,i}}\right)^{bs} \tag{92}$$

Comparing equations (91) and (92) shows that equation (91) approximates a straight line on Weibull probability paper. The straight relation of equation (92) is fit to the more exact relation of equation (91) numerically using linear regression. The reliability range used for fitting is 0.5 to 0.95, although wider ranges may also be used. Figure 42 is a two-parameter Weibull graph of these two relations for the spiral-bevel example of Appendix B.

The slope of the line found by regression is the drive system Weibull slope bs, and the life at which the drive system reliability, R_s, is ninety-percent is the $\ell_{10,s}$ life.

The ninety-percent reliability life is an important characteristic in design. This life for each component as well as for the transmission is reported by program TLIFE in both million output rotations and hours. The life in million output rotations, ℓ_0, can be expressed in terms of the ℓ_{10} life in hours as:

$$\ell_0 = \frac{60}{10^6} \cdot \omega_0 \cdot \ell_{10} \tag{93}$$

where ω_0 is the output speed in RPM.

Another important property in the estimation of service time is the mean time between overhauls. When the transmission is repaired with complete replacement, the service time is given by the mean transmission life in hours

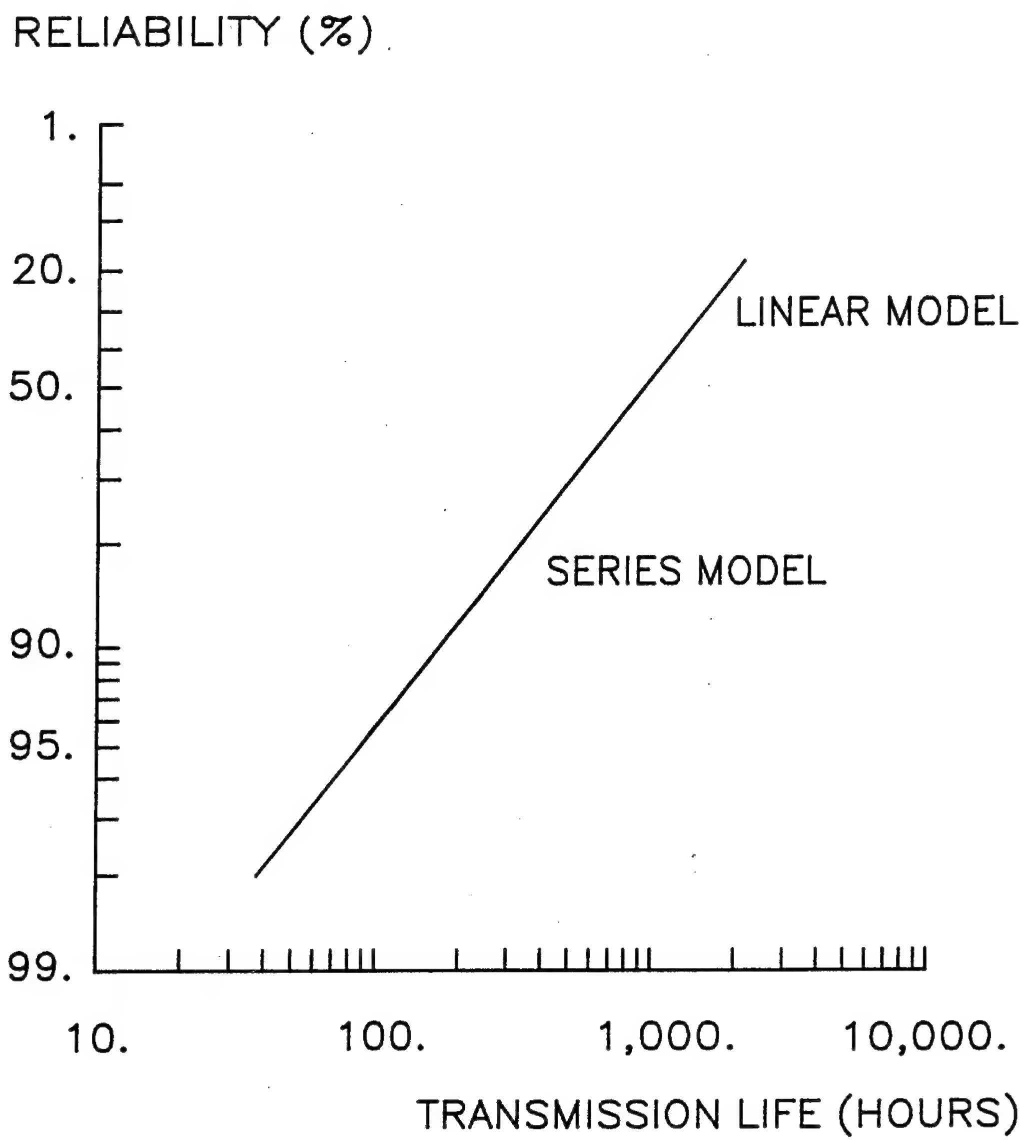


FIGURE 42 – SYSTEM RELIABILITY–LIFE CURVE FOR SPIRAL–BEVEL EXAMPLE

[21]. The mean life for a two-parameter Weibull distribution is expressed in terms of the gamma function as:

$$\ell_{av} = \theta \cdot \Gamma\,(1 + 1/b) \tag{94}$$

In terms of its ℓ_{10} life, the mean life can be expressed as:

$$\ell_{av} = \frac{\ell_{10} \cdot \Gamma\,(1 + 1/b)}{[\mathrm{Ln}(1/0.9)]^{1/b}} \tag{95}$$

Subroutine GAMMA evaluates the gamma function, $\Gamma\,(1 + 1/b)$.

If the repairs are component repairs, rather than full replacements, then the mean life between overhauls is based on a different statistical model [21]. A second mean life, which is the average of the mean lives of the individual components is estimated. In this case, the transmission failure rate, where failure rate is defined as the reciprocal of the mean life, is the sum of the individual component failure rates, and is given as:

$$\frac{1}{\ell_{av,s}} = \sum \frac{1}{\ell_{av,i}} \tag{96}$$

From this, the transmission mean life is estimated as the reciprocal of the failure rate:

$$\ell_{av,s} = \frac{1}{\sum 1/\ell_{av,i}} \tag{97}$$

This second mean transmission life is reported as the mean component life in the output summary. The system life is calculated in subroutine LIFE, using only the component lives and the Weibull slopes.

System Dynamic Capacities

The analysis for the drive system dynamic capacity is similar to the life analysis. The basic dynamic capacity is defined to be the drive system output torque, D_s, that produces a ninety-percent reliability drive system life of one-million output rotations. Equation (91), for these conditions can be written as:

$$1 = \sum \left(\frac{\ell_s}{\ell_{10,i}} \right)^{bi} \tag{98}$$

The ratio of the component load to the output torque is a constant for each component at all load levels in a drive system. The dynamic capacity of a component is for one-million component rotations. To calculate the drive system dynamic capacity one needs a similar common base which is chosen to be the output torque. This requires two conversions: the first to convert the component rotations to output shaft rotations and the second to convert the component load to output torque. The first conversion is:

$$C_{oi} = C_i \left(\frac{\omega_o}{\omega_i} \right)^{1/pi} \tag{99}$$

and the second is:

$$D_i = C_{oi} \frac{T_o}{F_i} \tag{100}$$

Replacing the actual and basic dynamic component loads in equation (79) with the corresponding drive system output torques, we have:

$$\ell_{10,i} = \left(\frac{D_i}{T} \right)^{pi} \tag{101}$$

Where D_i is the component dynamic capacity in units of output torque, $\ell_{10,i}$ is the ninety-percent reliability life of the component, and T is the drive system output torque which produces this ninety-percent reliability life. By definition, if the ninety-percent drive-system life is one-million output rotations, the output torque, T, will be the corresponding drive system dynamic capacity, D_s. Replacing T by D_s in equation (101), and substituting in equation (98) with ℓ_s equal to one million output rotations, gives:

$$1 = \Sigma \left(\frac{D_s}{D_i} \right)^{bi \cdot pi} \tag{102}$$

This again is not a simple load-life relationship. The load-life exponents for all components must be equal for a constant slope load-life relationship, which is:

$$\ell_{10,s} = \left(\frac{D_s}{T} \right)^{ps} \tag{103}$$

Figure 43 is a log-load versus log-life graph of the exact relationship of equation (102) and the approximate relationship of equation (103) for the spiral-bevel example of Appendix B.

Equation (102) is solved numerically for D_s with a starting value of the lowest component dynamic capacity, D_i. From this system dynamic capacity, D_s, the system load-life exponent may be calculated from equation (103) as:

$$ps = \frac{\mathrm{Ln}\ \ell_{10,s}}{\mathrm{Ln}\ (D_s/T)} \tag{104}$$

When a component load approaches the system dynamic capacity, the straight line fit to the curve approaches a tangent and the evaluation of the

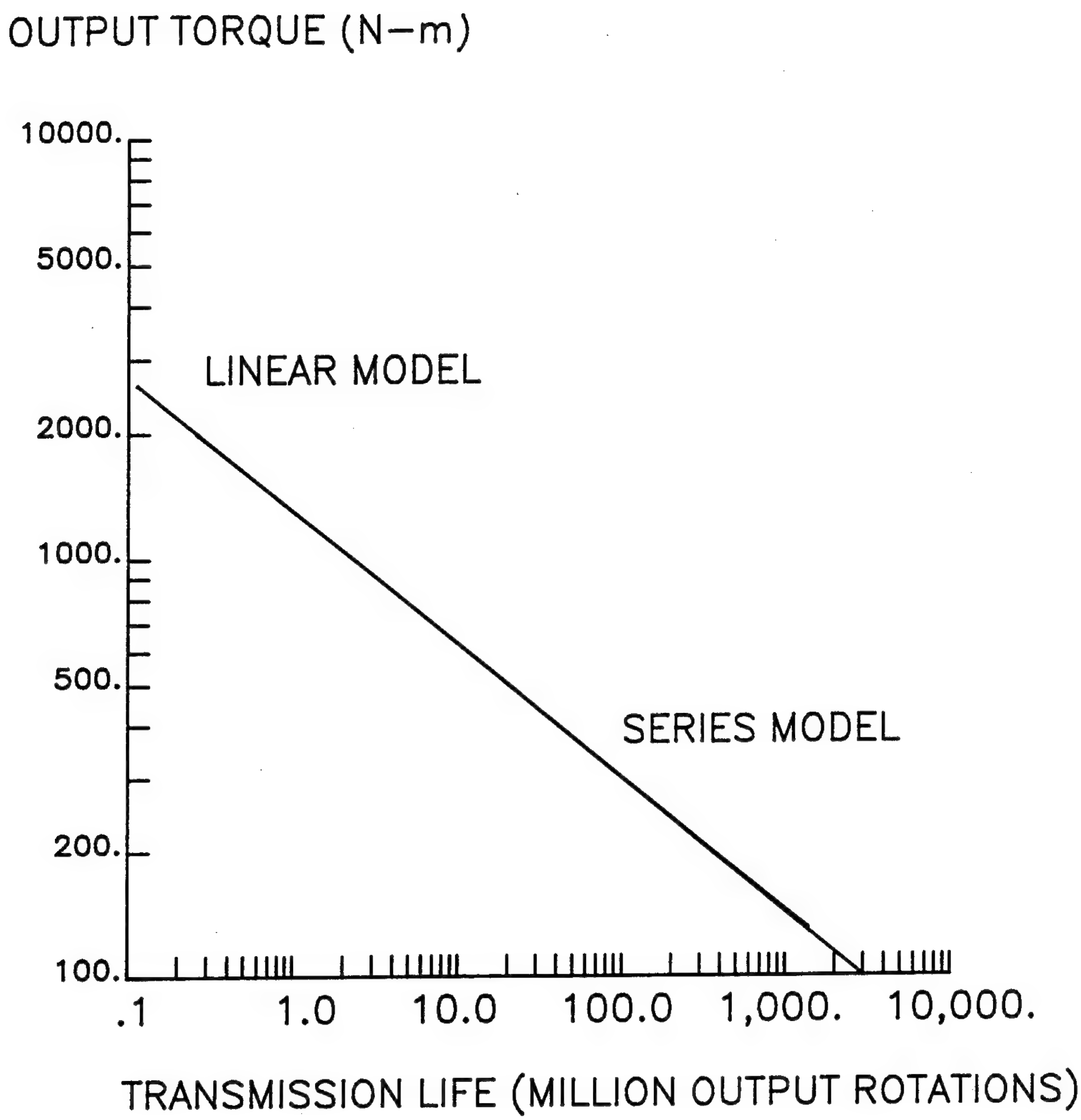


FIGURE 43 – SYSTEM LOAD–LIFE CURVE
FOR SPIRAL–BEVEL EXAMPLE

denominator in equation (104) will be difficult. When the output torque is within ten percent of the system dynamic capacity, the system load-life exponent is calculated from the weighted average of the component load-life factors. The weighting is done by the sum of the reciprocals of the component dynamic capacities to favor the weak components. The system load-life exponent for this case is calculated as:

$$ps = \frac{\sum pi/D_i}{\sum 1/D_i} \tag{105}$$

Subroutine DYN performs these calculations using this algorithm.

DISCUSSION OF RESULTS

This report describes the use of a Fortran program, TLIFE, which provides a generalized life and reliability model for the service life of transmissions composed of: spur, helical and spiral-bevel gears with ball, straight-roller and tapered-roller bearings. This program can analyze a variety of single and multiple-reduction transmissions for component and system service life. It is an upgrade of the transmission life analysis programs CHOPPER [8] and PSHAFT [10]. TLIFE can analyze all cases covered by these programs and many more.

The program enables the evaluation of transmission service life before the transmission is constructed. With this model, one can optimize a transmission to maximize its system service life while maintaining given power and weight constraints.

The transmissions that may be analyzed include: spur, helical and spiral-bevel reductions as well as series combinations of these reductions. The basic spur and helical reductions include: single-mesh, compound, parallel path, reverted star and planetary gear trains and single-plane spur star and planetary reductions. The spiral-bevel reductions include single-mesh and dual-input, single-reduction drives with arbitrary shaft angles. Up to twenty-five unit transmissions may be included in a transmission with the limitation that only one dual-input spiral-bevel be included and that its two inputs be identical. Both straddle and overhung bearing support for all gears are possible as is the use of a ring gear for the output.

The life model for this analysis is the two-parameter Weibull failure distribution, which has been applied to the bearings and gears of the transmissions as well as the overall transmission systems.

The Fortran program, TLIFE, which performs these analyses has been described and its use illustrated with three design examples. Extensive descriptions of the input file generation are provided in the PROGRAM USE section of this report, in the examples and in an ASCII '.DOC' file which is appended to the report and included at the start of the Fortran source file, 'TLIFE.FOR.'

A spiral-bevel single-mesh transmission was analyzed for its service life at a given power level as specified by its input torque and speed. A compound helical-gear transmission was analyzed first as a single compound reduction and again as two single-mesh reductions in series to illustrate the flexibility of the program. This transmission was analyzed for its service life at a specified power level. The separate service lives of the two component parts of this transmission were presented as well. Both analyses of the compound helical-gear transmission produced the same results.

Structurally, the program works with a main property array which serves as a property database. This database and the program structure could enable one to enter a series of component properties and determine a system life from the table and system subroutines alone by modifying the program. Due to extensive use of subroutines and modular programming techniques, it can expand easily to analyze other transmission configurations. The program is written in Fortran 77 and has been executed both in the personal computer DOS environment and on UNIX work stations. The analysis may be performed in either the SI metric or the English inch system of units.

In performing the life analyses, the program calculates the dynamic capacity of the transmission as that torque value with which one-million output rotations can be obtained with a ninety-percent reliability. It also

calculates the ninety-percent reliability life of the transmission at a given input torque and speed in million output rotations and in hours.

For a transmission life summary, the program calculates two mean lives for the transmission which represent the average time between overhauls for the transmission. The average lives are for either full transmission replacement at repair or failed component replacement only. The first mean life is the transmission mean life, while the second mean life is the overall component mean life.

SUMMARY OF RESULTS

This report describes a computer simulation program, "TLIFE," which models the service life of a transmission. The program is written in ANSI standard Fortran 77 and has an executable file of about 157 K bytes for use on a personal computer running DOS. It can also be compiled and executed in the UNIX computing environment. The computer program analyzes any one of eleven unit transmissions either singly or in a series combination of up to twenty five unit transmissions. Metric or English unit calculations are performed with the same routines using consistent input data and a flag to indicate whether the gear module or diametral pitch formulas are to be used.

The unit transmissions analyzed in the program are:

1. The single-mesh spur-gear reduction,
2. The single-mesh helical-gear reduction,
3. The compound spur-gear reduction,
4. The compound helical-gear reduction,
5. The parallel spur-gear reduction,
6. The parallel helical-gear reduction,
7. The reverted spur-gear reduction,
8. The reverted helical-gear reduction,
9. The single-plane spur-gear reduction,
10. The spiral-bevel reduction, and
11. The dual spiral-bevel reduction.

All options may have a final ring gear. The input and output gears may be supported by two bearings in straddle or overhung mountings. Four options exist for the support geometry of the intermediate shafts:

1) double straddle,

2) double overhung,

3) overhung output gear, and

4) overhung input gear.

The program runs in the personal computer environment with minimal user keyboard input required during execution. It prompts the user for the data file name prefix. It then reads the input ASCII data file, runs the analysis and writes the results of its analysis to a second data file with the same prefix and a ".OUT" extension.

The program uses a modular approach to separate the component calculations from the system calculations. It also uses a property array as a database in the calculations to maintain generality in the transmission dynamic capacity and life calculations.

This report describes the program and its input and output data files. An extensive description of the use of the program is included to facilitate its application. To illustrate its use, three examples are presented which demonstrate the information it can provide. The report also includes a development of the theory behind the analysis in the program.

APPENDIX A

TLIFE.DOC FILE

```
C
C     PROGRAM:  TLIFE
C
C          MAINTENANCE LIFE, DYNAMIC CAPACITY AND
C          RELIABILITY SIMULATION OF TRANSMISSIONS
C
C          THE TRANSMISSION MAY BE A SINGLE UNIT TRANSMISSION
C          OR A SERIES OF UP TO TWENTY-FIVE UNIT TRANSMISSIONS
C          IN A TOTAL SYSTEM TRANSMISSION.
C
C          THIS FILE IS INCLUDED AT THE START OF THE SOURCE LISTIING
C          OF THE PROGRAM.
C
C     REFERENCES:
C          SAVAGE, M., PRASANNA, M. G., AND RUBADEUX, K. L.; "TLIFE -
C          A PROGRAM FOR SPUR, HELICAL AND SPIRAL BEVEL TRANSMISSION
C          LIFE AND RELIABILITY MODELING," NASA CR-XXXX,
C          GRANT NAG 3-1047, JULY, 1994.
C
C          SAVAGE, M.; "LIFE AND DYNAMIC CAPACITY MODELING FOR
C          AIRCRAFT TRANSMISSIONS," NASA CR-4341, GRANT NAG 3-55,
C          JANUARY, 1991.
C
C          SAVAGE, M., RADIL, K.C., LEWICKI, D.G., AND COY, J.J.:
C          "COMPUTERIZED LIFE AND RELIABILITY MODELING FOR
C          TURBOPROP TRANSMISSIONS", JOURNAL OF PROPULSION AND
C          POWER, VOL. 5, NO. 5, SEPT.-OCT. 1989, PP. 610-614
C          (ALSO NASA TM-200918, AVSCOM TR-87-C-37,
C          AIAA PAPER NO. AIAA-88-2979, 1988).
C
C     REV. 1, 05/01/88 WRITTEN ON PERSONAL COMPUTER BY:  M. SAVAGE &
C         K.C. RADIL, USING A ZENITH Z-200 SERIES PERSONAL COMPUTER,
C         DOS 3.2 AND MICROSOFT FORTRAN 4.01 AT THE UNIVERSITY OF
C         AKRON, SUPPORTED BY NASA GRANT NAG 3-55.
C
C     REV. 2, 03/01/89, IMPLEMENTED ON NASA COMPUTER BY D.G. LEWICKI
C         USING A ZENITH Z-200 SERIES PERSONAL COMPUTER
C         (IBM AT COMPATIBLE),DOS 3.20 OPERATING SYSTEM,
C         AND MICROSOFT OPTIMIZING COMPILER VERSION 4.01.
C
C     REV. 3, 07/01/90, IMPLEMENTED ON PERSONAL COMPUTER BY M. SAVAGE
C         USING A ZENITH Z-200 SERIES PERSONAL COMPUTER
C         (IBM AT COMPATIBLE), DOS 3.3+ OPERATING SYSTEM,
C         AND MICROSOFT OPTIMIZING COMPILER VERSION 5.0.
C
C     REV. 4, 05/01/92, IMPLEMENTED ON PERSONAL COMPUTER BY M. PRASANNA
C         USING A ZENITH Z-200 SERIES PERSONAL COMPUTER
C         (IBM AT COMPATIBLE), DOS 4.0+ OPERATING SYSTEM,
```

```
C          AND MICROSOFT OPTIMIZING COMPILER VERSION 5.0.
C          AND IN UNIX ENVIRONMENT ON HP WORK STATIONS
C          WITH THE HP FORTRAN77 COMPILER.
C
C     REV. 5, 10/01/93, IMPLEMENTED ON PERSONAL COMPUTER BY M. SAVAGE
C          USING A GATEWAY 2000 SERIES 486 PERSONAL COMPUTER
C          (IBM AT COMPATIBLE), DOS 5.0 OPERATING SYSTEM,
C          AND MICROSOFT OPTIMIZING COMPILER VERSION 5.1.
C
C           TO RUN PROGRAM, TYPE PROGRAM NAME.
C           AT THE PROMPT, ENTER DATA FILE PREFIX.
C           INPUT DATA FILE SHOULD HAVE A ".in" EXTENSION.
C           THE OUTPUT DATA FILE WILL HAVE THE SAME PREFIX,
C           AND A ".out" EXTENSION.
C
C       T L I F E   I N P U T   D A T A   D E S C R I P T I O N
C       =========   =========   =======   =====================
C
C       (FORMAT FREE INPUT, USE BLANKS OR COMMAS AS SEPARATORS)
C             (ALL VARIABLES ARE REAL EXCEPT WHERE NOTED)
C           (EACH LINE OF DATA SHOULD BE KEPT ON A SINGLE LINE)
C
C  LINE/
C  VARIABLE      DESCRIPTION
C
C  LINE #0      TRANSMISSION SYSTEM DATA LINE, GROUP #A
C
C           THEN FOR EACH UNIT TRANSMISSION
C
C  LINE #1      TRANSMISSION TYPE, GROUP #B
C            =====================================================
C              FOR SINGLE MESH SPUR GEAR REDUCTION (LINE #1 = 1)
C            FOR SINGLE MESH HELICAL GEAR REDUCTION (LINE #1 = 2)
C            FOR SINGLE MESH SPIRAL BEVEL REDUCTION (LINE #1 = 10)
C            FOR DUAL MESH SPIRAL BEVEL REDUCTION (LINE #1 = 11)
C
C  LINE #2      TRANSMISSION DATA LINE, GROUP #C
C  LINE #3      MESH CHARACTERISTICS LINE, GROUP #D
C  LINE #4      GEAR DATA LINE, GROUP #E, FOR PINION
C  LINE #5      BEARING MOUNTING LINE, GROUP #F, FOR PINION SHAFT
C  LINE #6      BEARING DATA LINE, GROUP #G, FOR PINION BEARING #1
C  LINE #7      BEARING DATA LINE, GROUP #G, FOR PINION BEARING #2
C  LINE #8      GEAR DATA LINE, GROUP #E, FOR OUTPUT GEAR
C  LINE #9      BEARING MOUNTING LINE, GROUP #F, OUTPUT GEAR SHAFT
C  LINE #10     BEARING DATA LINE, GROUP #G, FOR OUTPUT GEAR BEARING #1
C  LINE #11     BEARING DATA LINE, GROUP #G, FOR OUTPUT GEAR BEARING #2
C
C  NOTE 1:  FOR PARALLEL COMPOUND REDUCTIONS, REVERTED REDUCTIONS
C            AND SINGLE PLANE REDUCTIONS, IT IS ASSUMED THAT
C            ALL INTERMEDIATE SHAFT/GEAR/BEARING ASSEMBLIES
C            ARE IDENTICAL.
```

```
C     ==============================================================
C      FOR SPUR GEAR COMPOUND REDUCTION (LINE #1 = 3)
C      FOR HELICAL GEAR COMPOUND REDUCTION (LINE #1 = 4)
C      FOR SPUR GEAR PARALLEL COMPOUND REDUCTION (LINE #1 = 5)
C      FOR HELICAL GEAR PARALLEL COMPOUND REDUCTION (LINE #1 = 6)
C      FOR REVERTED HELICAL GEAR REDUCTION (LINE #1 = 8)
C
C  LINE #2      TRANSMISSION DATA LINE, GROUP #C
C  LINE #3      MESH CHARACTERISTICS LINE, GROUP #D,
C                 FOR PINION-INTERMEDIATE GEAR MESH
C  LINE #4      MESH CHARACTERISTICS LINE, GROUP #D,
C                 FOR INTERMEDIATE-OUTPUT GEAR MESH
C  LINE #5      GEAR DATA LINE, GROUP #E, FOR PINION
C  LINE #6      BEARING MOUNTING LINE, GROUP #F, FOR PINION SHAFT
C  LINE #7      BEARING DATA LINE, GROUP #G, FOR PINION BEARING #1
C  LINE #8      BEARING DATA LINE, GROUP #G, FOR PINION BEARING #2
C  LINE #9      GEAR DATA LINE, GROUP #E,
C                 FOR INTERMEDIATE GEAR WHICH MESHES WITH PINION GEAR
C  LINE #10     GEAR DATA LINE, GROUP #E,
C                 FOR INTERMEDIATE GEAR WHICH MESHES WITH OUTPUT GEAR
C  LINE #11     BEARING MOUNTING LINE, GROUP #F, FOR INTERMEDIATE SHAFT
C  LINE #12     BEARING DATA LINE, GROUP #G,
C                 FOR INTERMEDIATE SHAFT BEARING #1
C  LINE #13     BEARING DATA LINE, GROUP #G,
C                 FOR INTERMEDIATE SHAFT BEARING #2
C  LINE #14     GEAR DATA LINE, GROUP #E, FOR OUTPUT GEAR
C  LINE #15     BEARING MOUNTING LINE, GROUP #F, OUTPUT GEAR SHAFT
C  LINE #16     BEARING DATA LINE, GROUP #G, FOR OUTPUT GEAR BEARING #1
C  LINE #17     BEARING DATA LINE, GROUP #G, FOR OUTPUT GEAR BEARING #2
C            ============================================
C                  FOR REVERTED REDUCTION (LINE #1 = 7)
C
C  LINE #2      TRANSMISSION DATA LINE, GROUP #C
C  LINE #3      MESH CHARACTERISTICS LINE, GROUP #D,
C                 FOR SUN GEAR-PLANET GEAR MESH
C  LINE #4      MESH CHARACTERISTICS LINE, GROUP #D,
C                 FOR PLANET GEAR-RING GEAR MESH
C  LINE #5      GEAR DATA LINE, GROUP #E, FOR SUN GEAR
C  LINE #6      GEAR DATA LINE, GROUP #E,
C                 FOR PLANET GEAR (WHICH MESHES WITH THE SUN GEAR)
C  LINE #7      GEAR DATA LINE, GROUP #E,
C                 FOR PLANET GEAR (WHICH MESHES WITH THE RING GEAR)
C  LINE #8      BEARING MOUNTING LINE, GROUP #F, FOR INTERMEDIATE SHAFT
C  LINE #9      BEARING DATA LINE, GROUP #G,
C                 FOR INTERMEDIATE SHAFT BEARING #1
C  LINE #10     BEARING DATA LINE, GROUP #G,
C                 FOR INTERMEDIATE SHAFT BEARING #2
C  LINE #11     GEAR DATA LINE, GROUP #E, FOR OUTPUT GEAR
```

```
C                   ============================================
C                   FOR SINGLE PLANE SPUR REDUCTION (LINE #1 = 9)
C
C  LINE #2       TRANSMISSION DATA LINE, GROUP #C
C  LINE #3       MESH CHARACTERISTICS LINE, GROUP #D,
C                  FOR SUN GEAR-PLANET GEAR MESH
C  LINE #4       MESH CHARACTERISTICS LINE, GROUP #D,
C                  FOR PLANET GEAR-RING GEAR MESH
C  LINE #5       GEAR DATA LINE, GROUP #E, FOR SUN GEAR
C  LINE #6       GEAR DATA LINE, GROUP #E,
C                  FOR PLANET GEAR (WHICH MESHES WITH THE SUN GEAR)
C  LINE #7       GEAR DATA LINE, GROUP #E,
C                  FOR PLANET GEAR (WHICH MESHES WITH THE RING GEAR)
C  LINE #8       BEARING DATA LINE, GROUP #G, FOR PLANET BEARINGS
C  LINE #9       GEAR DATA LINE, GROUP #E, FOR RING GEAR
C
C  NOTE 2:  WHEN TWO CONSECUTIVE UNIT TRANSMISSIONS WOULD HAVE TWO
C     OUTPUT GEAR BEARINGS FOR THE HIGH-SPEED UNIT AND TWO INPUT PINION
C     BEARINGS FOR THE LOW-SPEED UNIT, IT IS ASSUMED THAT THESE
C     BEARINGS ARE THE SAME AND THE THEY SUPPORT BOTH THE OUTPUT GEAR
C     OF THE HIGH-SPEED UNIT TRANSMISSION AND THE INPUT PINION OF THE
C     LOW-SPEED UNIT TRANSMISSION.  THE BEARINGS SHOULD BE ENTERED
C     TWICE, ONCE FOR EACH UNIT TRANSMISSION AS WELL AS A SHAFT ANGLE,
C     BETA, BETWEEN THE RADIAL FORCE ON THE OUTPUT GEAR OF THE HIGH-
C     SPEED UNIT AND THE RADIAL FORCE ON THE INPUT GEAR OF THE LOW-
C     SPEED UNIT.  THE BEARINGS SHOULD BE IDENTICAL IN TYPE, LOCATION,
C     CAPACITY AND THRUST LOAD DESIGNATION FOR THE TWO UNIT
C     TRANSMISSIONS.  THE PROGRAM WILL DETERMINE THE LOADS ON THE
C     BEARINGS FROM EACH UNIT TRANSMISSION AND SUPERIMPOSE THEM ON THE
C     INPUT BEARINGS OF THE LOW-SPEED UNIT.  THE OUTPUT BEARINGS OF
C     THE HIGH-SPEED UNIT WILL NOT BE COUNTED IN THE LIFE CALCULATIONS.
C  ================================================================
C
C                    GROUP #A - SYSTEM CHARACTERISTICS LINE
C                      (ENTER ALL FIVE VARIABLES)
C                      (THE FIRST TWO AND THE LAST ARE INTEGERS)
C
C  VARIABLE #A1  NUMT - NUMBER OF UNIT TRANSMISSIONS IN SYSTEM
C                       WITH NO DUAL BEVEL UNIT OR THE NUMBER OF UNIT
C                       TRANSMISSIONS IN ONE COMPLETE BRANCH INCLUDING
C                       OUTPUT UNITS FROM SYSTEM INPUT TO SYSTEM OUTPUT
C                       FOR A SYSTEM WITH ONE DUAL BEVEL UNIT
C  VARIABLE #A2  METRIC / ENGLISH UNIT FLAG
C                          MET = 1  - METRIC SI UNITS
C                          MET = 2  - ENGLISH INCH UNITS
C  VARIABLE #A3  INPUT TORQUE (kN - m) OR (LB - IN)
C  VARIABLE #A4  INPUT SPEED (RPM)
C  VARIABLE #A5  DIRECTION OF INPUT TORQUE AND SPEED (INTEGER)
C                 DIRECTION IS TAKEN AS LOOKING INTO THE TRANSMISSION
C                 AT THE INPUT SHAFT
C                         NDIR = 1 FOR COUNTER CLOCKWISE ROTATION
C                         NDIR = 2 FOR CLOCKWISE ROTATION
```

```
C
C                   GROUP #B - UNIT TRANSMISSION NUMBER
C
C  VARIABLE #B1  N - AN INTEGER DESCRIBING THE TYPE OF TRANSMISSION:
C           =  1  FOR SINGLE MESH SPUR REDUCTION
C           =  2  FOR SINGLE MESH HELICAL REDUCTION
C           =  3  FOR COMPOUND SPUR REDUCTION
C           =  4  FOR COMPOUND HELICAL REDUCTION
C           =  5  FOR PARALLEL COMPOUND SPUR REDUCTION
C           =  6  FOR PARALLEL COMPOUND HELICAL REDUCTION
C           =  7  FOR REVERTED SPUR REDUCTION (STAR OR PLANETARY)
C           =  8  FOR REVERTED HELICAL REDUCTION (STAR OR PLANETARY)
C           =  9  FOR SINGLE PLANE SPUR REDUCTION (STAR OR PLANETARY)
C           = 10  FOR SINGLE MESH SPIRAL BEVEL REDUCTION
C           = 11  FOR DUAL INPUT SPIRAL BEVEL REDUCTION
C
C
C                   GROUP #C - UNIT TRANSMISSION CONFIGURATION
C
C  NOTE 3:  DIFFERENT UNITS REQUIRE DIFFERENT ITEMS:
C                SINGLE UNIT SPUR AND HELICAL REDUCTIONS - #C1
C     COMPOUND SPUR AND HELICAL REDUCTIONS - #C1,#C5
C     PARALLEL SPUR AND HELICAL REDUCTIONS - #C1,#C4,#C5
C                    SINGLE UNIT SPIRAL BEVEL REDUCTIONS - #C1,#C5
C                           DUAL SPIRAL BEVEL REDUCTIONS - #C1,#C5,#C6
C                   SPUR AND HELICAL REVERTED REDUCTIONS - #C1,#C2,#C3
C                           SINGLE PLANE SPUR REDUCTIONS - #C1,#C2,#C3
C     ANY UNIT THAT FOLLOWS A UNIT AND SHARES BEARINGS
C     REQUIRES THE ADDITION OF #C7 TO THIS DATA LINE.
C
C  VARIABLE #C1 NRING - AN INTEGER DESCRIBING THE FINAL GEAR
C                       NRING = 1 - EXTERNAL FINAL GEAR
C                       NRING = 2 - INTERNAL FINAL GEAR (RING)
C  VARIABLE #C2  NP - NUMBER OF PARALLEL LOAD PATHS OR PLANETS
C  VARIABLE #C3  NARM - TRANSMISSION OUTPUT OPTION
C                       NARM  = 1 - FINAL GEAR IS OUTPUT, ARM IS FIXED
C                       NARM  = 2 - FINAL GEAR IS FIXED, ARM IS OUTPUT
C  VARIABLE #C4  NOUT - PARALLEL COMPOUND OUTPUT SHAFT LOCATION
C                       NOUT  = 1 - OUTPUT GEAR ON SAME SIDE OF
C                                   INTERMEDIATE SHAFTS AS INPUT SHAFT
C                       NOUT  = 2 - OUTPUT GEAR ON OPPOSITE SIDE OF
C                                   INTERMEDIATE SHAFTS FROM INPUT SHAFT
C  VARIABLE #C5  SIGMA - SHAFT ANGLE (DEGREES)
C                       COUNTER CLOCKWISE FROM INPUT TO OUTPUT
C                       ABOUT INTERMEDIATE SHAFT AS VIEWED FROM INPUT
C                       FOR COMPOUND REDUCTIONS
C                       OR THE ANGLE BETWEEN THE LOCATIONS OF THE TWO
C                       INTERMEDIATE SHAFTS RELATIVE TO THE INPUT
C                       SHAFT FOR PARALLEL REDUCTIONS
C                       OR THE ANGLE FROM THE BACK SIDE OF THE INPUT
C                       BEVEL GEAR TO THE BACK SIDE OF THE OUTPUT
C                       BEVEL GEAR FOR SPIRAL BEVEL REDUCTIONS
```

```
C  VARIABLE #C6  LAMBDA - ANGLE BETWEEN INPUT PINIONS (DEGREES)
C                       FOR DUAL SPIRAL REDUCTION
C  VARIABLE #C7  BETA - ANGLE BETWEEN OUTPUT AND INPUT RADIAL
C                       GEAR FORCES (DEGREES) FOR COUPLING UNIT
C                       TRANSMISSIONS - MEASURED COUNTER-CLOCKWISE FROM
C                       FIRST UNIT'S OUTPUT GEAR FORCE TO THE SECOND
C                       UNITS INPUT GEAR FORCE ABOUT THE INPUT SHAFT
C                       LOOKING INTO THE SECOND UNIT FROM THE FIRST.
C                       FOR PARALLEL COMPOUND UNITS AND DUAL BEVEL
C                       OUTPUT GEARS, THE RESULTANT FORCE DIRECTION
C                       LIES MIDWAY BETWEEN THE TWO MESH CONTACT POINTS.
C
C
C                GROUP #D - MESH CHARACTERISTICS LINE
C
C  NOTE 4:  VARIABLES #D4 AND #D5 ARE REQUIRED ONLY FOR
C           HELICAL AND SPIRAL BEVEL UNIT TRANSMISSIONS.
C
C  VARIABLE #D1  NORMAL GEAR MESH MODULE (mm) OR NORMAL DIAMETRAL
C                       PITCH (TEETH/IN) FOR SPUR OR HELICAL GEARS, OR
C                BACK CONE DISTANCE OF THE GEAR MESH (mm) OR (IN)
C                       FOR SPIRAL BEVEL GEARS
C  VARIABLE #D2  NORMAL PRESSURE ANGLE OF THE MESH (DEG)
C  VARIABLE #D3  AXIAL FACE WIDTH OF MESH (mm) OR (IN)
C  VARIABLE #D4  HELIX OR SPIRAL ANGLE OF THE GEAR MESH (DEG)
C  VARIABLE #D5  HELIX OR SPIRAL HAND OF THE DRIVING PINION (INTEGER)
C                       = 1 FOR RIGHT HAND
C                       = 2 FOR LEFT HAND
C                       = 3 FOR HERRINGBONE GEARS
C
C
C                     GROUP #E - GEAR DATA LINE
C
C  VARIABLE #E1  NUMBER OF TEETH ON GEAR
C  VARIABLE #E2  ADDENDUM OF GEAR (mm) OR (IN)
C                    MEASURED IN BACK PLANE FOR SPIRAL BEVEL GEARS
C  VARIABLE #E2  GEAR WEIBULL EXPONENT
C  VARIABLE #E3  GEAR SURFACE STRENGTH (MPa) OR (KSI)
C  VARIABLE #E4  GEAR LOAD-LIFE EXPONENT
C
C
C                  GROUP #F - BEARING MOUNTING LINE
C
C  VARIABLE #F1  BEARING TAKING THRUST LOAD (INTEGER)
C                 = 0 FOR NO THRUST LOADS ON EITHER BEARING
C                 = 1 FOR BEARING #1 TAKING THE THRUST LOAD
C                 = 2 FOR BEARING #2 TAKING THE THRUST LOAD
C                 = 3 FOR BOTH BEARINGS SHARING THE THRUST LOAD EQUALLY
C  VARIABLE #F2  TYPE OF BEARING MOUNTING (INTEGER):
C                 FOR A SINGLE GEAR SHAFT
C                  = 1 FOR CASE #1 (STRADDLE MOUNTING)
C                  = 2 FOR CASE #2 (OVERHUNG MOUNTING), OR
```

```
C                    FOR A SHAFT SUPPORTING TWO GEARS
C                     = 1 FOR CASE #1 - DOUBLE STRADDLE
C                     = 2 FOR CASE #2 - DOUBLE OVERHUNG
C                     = 3 FOR CASE #3 - STRADDLE GEAR #1, OVERHANG GEAR #2
C                     = 4 FOR CASE #4 - OVERHANG GEAR #1, STRADDLE GEAR #2
C  VARIABLE #F3  DISTANCE A FOR SINGLE GEAR SUPPORT (mm) OR (IN), OR
C                DISTANCE C FOR TWO GEAR SUPPORT (mm) OR (IN)
C  VARIABLE #F4  DISTANCE B FOR SINGLE GEAR SUPPORT (mm) OR (IN), OR
C                DISTANCE D FOR TWO GEAR SUPPORT (mm) OR (IN)
C  VARIABLE #F5  DISTANCE E FOR TWO GEAR SUPPORT (mm) OR (IN)
C
C          SINGLE GEAR SUPPORT CONFIGURATIONS
C
C    CASE #1:
C                     BRG#1=======GEAR=======BRG#2
C                       <----A-----><----B----->
C
C    CASE #2:
C                     GEAR=======BRG#1=======BRG#2
C                       <----A----->
C                       <----------B----------->
C
C
C          TWO GEAR SUPPORT CONFIGURATIONS
C
C    CASE #1:
C                     BRG#1======GEAR#1=====GEAR#2=====BRG#2
C                       <----C-----><----D----><----E---->
C
C    CASE #2:
C                     GEAR#1=====BRG#1======BRG#2======GEAR#2
C                       <----C----->           <----E---->
C                       <---------------D--------------->
C
C    CASE #3:
C                     BRG#1======GEAR#1=====BRG#2======GEAR#2
C                       <----C----->           <----E---->
C                                   <--------D----------->
C
C    CASE #4:
C                     GEAR#1=====BRG#1======GEAR#2=====BRG#2
C                       <----C----->           <----E---->
C                       <----------D---------->
C
C    WHERE, GEAR#1 --> INTERMEDIATE GEAR WHICH MESHES WITH PINION
C           GEAR#2 --> INTERMEDIATE GEAR WHICH MESHES WITH OUTPUT GEAR
C
C
```

```
C                          GROUP #G - BEARING DATA LINE
C
C   NOTE 5: BALL AND ROLLER BEARINGS WHICH SEE ONLY RADIAL LOADS
C           REQUIRE ONLY THE FIRST FIVE ITEMS,
C           BALL BEARINGS WHICH SEE THRUST LOADS REQUIRE
C           THE FIRST SEVEN ITEMS, AND
C           TAPERED ROLLER BEARINGS WHICH SEE THRUST LOADS REQUIRE
C           THE FIRST FIVE AND THE LAST TWO ITEMS.
C
C   VARIABLE #G1  BEARING TYPE (INTEGER):
C                   = 1 FOR SINGLE ROW BALL BEARING
C                   = 2 FOR DOUBLE ROW BALL BEARING
C                   = 3 FOR SINGLE ROW ROLLER BEARING
C                   = 4 FOR DOUBLE ROW ROLLER BEARING
C                   = 5 FOR DOUBLE ROW TAPERED ROLLER BEARING
C   VARIABLE #G2  BASIC DYNAMIC CAPACITY OF BEARING (kN) OR (LBS)
C   VARIABLE #G3  BEARING WEIBULL EXPONENT
C   VARIABLE #G4  BEARING LIFE ADJUSTMENT FACTOR
C   VARIABLE #G5  BEARING RACE ROTATION FACTOR
C   VARIABLE #G6  STATIC CAPACITY OF BALL BEARING (kN) OR (LBS)
C   VARIABLE #G7  BALL BEARING CONTACT ANGLE (DEG)
C   VARIABLE #G8  TAPERED ROLLER AXIAL PRELOAD (kN) OR (LBS)
C   VARIABLE #G9  RADIAL TO AXIAL THRUST CAPACITY RATIO
C
C   NOTE 6: IF THE PROPERTY ARRAY SIZE IS INSUFFICIENT, THE TRANSMISSION
C           ANALYSIS IS DONE FOR THE TRANSMISSIONS WHICH FIT.
C
```

APPENDIX B

SPIRAL BEVEL EXAMPLE OUTPUT FILE

```
SPIRAL BEVEL GEAR REDUCTION RESULTS


 INPUT SPEED..............................       4000.00  RPM
 OUTPUT SPEED.............................       2000.00  RPM
 SPEED REDUCTION RATIO....................          2.00
 TRANSMITTED POWER........................        104.720 kW
 INPUT TORQUE.............................        250.000 N-m
 DIRECTION OF INPUT.......................     COUNTER CLOCKWISE
 OUTPUT TORQUE............................        500.000 N-m
 DIRECTION OF OUTPUT......................     CLOCKWISE



PINION CHARACTERISTICS AND MOUNTING

OVERHUNG MOUNTING

          GEAR=======BRG#1=======BRG#2
            <----A----->
            <----------B----------->

 DISTANCE A...............................         75.000 mm
 DISTANCE B...............................        200.000 mm

     BEARING #1

SINGLE ROW ROLLER BEARING

 BASIC DYNAMIC CAPACITY...................         19.700 kN
 ROTATION FACTOR..........................          1.00
 WEIBULL EXPONENT.........................          1.13
 LIFE ADJUSTMENT FACTOR...................          6.00

 RADIAL FORCE.............................         -1.337 kN
 TANGENTIAL FORCE.........................          7.726 kN
 TOTAL EQUIVALENT RADIAL FORCE............          7.841 kN
 ADJUSTED DYNAMIC CAPACITY................         27.482 kN
```

BEARING #2

SINGLE ROW BALL BEARING

STATIC CAPACITY...........................	35.000 kN
CONTACT ANGLE.............................	25.00 DEG
BASIC DYNAMIC CAPACITY....................	10.900 kN
ROTATION FACTOR...........................	1.00
WEIBULL EXPONENT..........................	1.11
LIFE ADJUSTMENT FACTOR....................	6.00

AXIAL FORCE...............................	3.999 kN
RADIAL FORCE..............................	1.537 kN
TANGENTIAL FORCE..........................	-2.897 kN
TOTAL EQUIVALENT RADIAL FORCE.............	4.824 kN
ADJUSTED DYNAMIC CAPACITY.................	15.721 kN

PINION

NUMBER OF TEETH...........................	21
MID CONE PITCH RADIUS.....................	51.773 mm
NORMAL MODULE.............................	4.04 mm
NORMAL PRESSURE ANGLE.....................	20.00 DEG
FACE WIDTH................................	30.000 mm
SHAFT ANGLE...............................	110.000 DEG
BACK CONE DISTANCE........................	120.000 mm
CONE ANGLE................................	29.543 DEG
SPIRAL ANGLE OF THE GEAR..................	35.000 DEG
SPIRAL DIRECTION..........................	RIGHT HAND
MID CONE REFERENCE SPUR GEAR RADIUS.......	88.688 mm
BACK CONE ADDENDUM........................	5.633 mm
DIRECTION OF GEAR ROTATION................	COUNTER CLOCKWISE
SURFACE STRENGTH..........................	1500.000 MPa
WEIBULL EXPONENT..........................	2.500
LOAD-LIFE FACTOR..........................	8.930

THE INVOLUTE CONTACT RATIO OF THE MESH IS.	1.148
THE FACE CONTACT RATIO OF THE MESH IS.....	1.357

FORCES

AXIAL FORCE...............................	3.999 kN
RADIAL FORCE..............................	0.199 kN
TANGENTIAL FORCE..........................	4.829 kN
TOTAL FORCE...............................	6.273 kN
ADJUSTED DYNAMIC CAPACITY.................	34.783 kN

OUTPUT GEAR CHARACTERISTICS AND MOUNTING

OVERHUNG MOUNTING

```
GEAR=======BRG#1=======BRG#2
  <----A----->
  <----------B----------->
```

DISTANCE A................................	75.000 mm
DISTANCE B................................	200.000 mm

BEARING #1

SINGLE ROW ROLLER BEARING

BASIC DYNAMIC CAPACITY....................	23.300 kN
ROTATION FACTOR...........................	1.00
WEIBULL EXPONENT..........................	1.13
LIFE ADJUSTMENT FACTOR....................	6.00
RADIAL FORCE..............................	4.616 kN
TANGENTIAL FORCE..........................	7.726 kN
TOTAL EQUIVALENT RADIAL FORCE.............	9.000 kN
ADJUSTED DYNAMIC CAPACITY.................	40.101 kN

BEARING #2

SINGLE ROW BALL BEARING

STATIC CAPACITY...........................	35.000 kN
CONTACT ANGLE.............................	25.00 DEG
BASIC DYNAMIC CAPACITY....................	12.900 kN
ROTATION FACTOR...........................	1.00
WEIBULL EXPONENT..........................	1.11
LIFE ADJUSTMENT FACTOR....................	6.00
AXIAL FORCE...............................	1.555 kN
RADIAL FORCE..............................	-0.926 kN
TANGENTIAL FORCE..........................	-2.897 kN
TOTAL EQUIVALENT RADIAL FORCE.............	3.042 kN
ADJUSTED DYNAMIC CAPACITY.................	23.441 kN

OUTPUT GEAR

NUMBER OF TEETH...........................	42
MID CONE PITCH RADIUS.....................	103.547 mm
NORMAL MODULE.............................	4.04 mm
NORMAL PRESSURE ANGLE.....................	20.00 DEG
FACE WIDTH................................	30.000 mm
SHAFT ANGLE...............................	110.000 DEG

BACK CONE DISTANCE........................	120.000 mm
CONE ANGLE................................	80.457 DEG
SPIRAL ANGLE OF THE GEAR..................	35.000 DEG
SPIRAL DIRECTION..........................	LEFT HAND
MID CONE REFERENCE SPUR GEAR RADIUS.......	930.774 mm
BACK CONE ADDENDUM........................	2.749 mm
DIRECTION OF GEAR ROTATION................	CLOCKWISE
SURFACE STRENGTH..........................	1500.000 MPa
WEIBULL EXPONENT..........................	2.500
LOAD-LIFE FACTOR..........................	8.930

FORCES

AXIAL FORCE...............................	1.555 kN
RADIAL FORCE..............................	3.690 kN
TANGENTIAL FORCE..........................	4.829 kN
TOTAL FORCE...............................	6.273 kN
ADJUSTED DYNAMIC CAPACITY.................	36.441 kN

COMPONENT AND TRANSMISSION

OUTPUT DYNAMIC CAPACITY AND LIFE

PINION		
DYNAMIC CAPACITY.........................	2772.356	N-m
LOAD-LIFE EXPONENT.......................	8.93	
WEIBULL EXPONENT.........................	2.50	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	4393803.	
L10 LIFE.................................	0.3661502E+08	HOURS
MEAN LIFE................................	0.8208946E+08	HOURS
PINION BEARING #1		
DYNAMIC CAPACITY.........................	1752.483	N-m
LOAD-LIFE EXPONENT.......................	3.30	
WEIBULL EXPONENT.........................	1.13	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	62.72724	
L10 LIFE.................................	522.7270	HOURS
MEAN LIFE................................	3429.324	HOURS
PINION BEARING #2		
DYNAMIC CAPACITY.........................	1629.355	N-m
LOAD-LIFE EXPONENT.......................	3.00	
WEIBULL EXPONENT.........................	1.11	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	34.60489	
L10 LIFE.................................	288.3741	HOURS
MEAN LIFE................................	1939.397	HOURS

OUTPUT GEAR

DYNAMIC CAPACITY........................	2904.525	N-m
LOAD-LIFE EXPONENT......................	8.93	
WEIBULL EXPONENT........................	2.50	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	6659760.	
L10 LIFE................................	0.5549800E+08	HOURS
MEAN LIFE...............................	0.1244244E+09	HOURS

OUTPUT GEAR BEARING #1

DYNAMIC CAPACITY........................	2227.919	N-m
LOAD-LIFE EXPONENT......................	3.30	
WEIBULL EXPONENT........................	1.13	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	138.5054	
L10 LIFE................................	1154.212	HOURS
MEAN LIFE...............................	7572.148	HOURS

OUTPUT GEAR BEARING #2

DYNAMIC CAPACITY........................	3853.475	N-m
LOAD-LIFE EXPONENT......................	3.00	
WEIBULL EXPONENT........................	1.11	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	457.7701	
L10 LIFE................................	3814.751	HOURS
MEAN LIFE...............................	25655.27	HOURS

TRANSMISSION

DYNAMIC CAPACITY........................	1313.668	N-m
LOAD-LIFE EXPONENT......................	3.13	
WEIBULL EXPONENT........................	1.12	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	20.65963	
L10 LIFE................................	172.1635	HOURS
MEAN LIFE...............................	1146.136	HOURS
MEAN COMPONENT LIFE.....................	1022.193	HOURS

APPENDIX C

COMPOUND HELICAL EXAMPLE OUTPUT FILE

COMPOUND HELICAL GEAR REDUCTION RESULTS

INPUT SPEED................................	5000.00 RPM
OUTPUT SPEED...............................	555.56 RPM
SPEED REDUCTION RATIO......................	9.00
TRANSMITTED POWER..........................	47.600 HP
INPUT TORQUE...............................	600.000 LB-IN
DIRECTION OF INPUT.........................	COUNTER CLOCKWISE
OUTPUT TORQUE..............................	5400.000 LB-IN
DIRECTION OF OUTPUT........................	COUNTER CLOCKWISE
NUMBER OF PARALLEL LOAD PATHS..............	1

PINION CHARACTERISTICS AND MOUNTING

OVERHUNG MOUNTING

```
GEAR=======BRG#1=======BRG#2
  <----A----->
  <----------B----------->
```

DISTANCE A.................................	3.000 IN
DISTANCE B.................................	8.000 IN

BEARING #1

SINGLE ROW ROLLER BEARING

BASIC DYNAMIC CAPACITY.....................	4070.000 LBS
ROTATION FACTOR............................	1.00
WEIBULL EXPONENT...........................	1.13
LIFE ADJUSTMENT FACTOR.....................	6.00
RADIAL FORCE...............................	210.247 LBS
TANGENTIAL FORCE...........................	665.107 LBS
TOTAL EQUIVALENT RADIAL FORCE..............	697.547 LBS
ADJUSTED DYNAMIC CAPACITY..................	3599.427 LBS

BEARING #2

SINGLE ROW BALL BEARING

STATIC CAPACITY............................	7880.000 LBS
CONTACT ANGLE..............................	25.00 DEG
BASIC DYNAMIC CAPACITY.....................	2248.000 LBS
ROTATION FACTOR............................	1.00
WEIBULL EXPONENT...........................	1.11
LIFE ADJUSTMENT FACTOR.....................	6.00
AXIAL FORCE................................	240.000 LBS
RADIAL FORCE...............................	-35.541 LBS
TANGENTIAL FORCE...........................	-249.415 LBS
TOTAL EQUIVALENT RADIAL FORCE..............	312.093 LBS
ADJUSTED DYNAMIC CAPACITY..................	1963.809 LBS

PINION

NUMBER OF TEETH............................	15
PITCH RADIUS...............................	1.443 IN
NORMAL PITCH...............................	6.00 1./IN
NORMAL PRESSURE ANGLE......................	20.00 DEG
FACE WIDTH.................................	0.750 IN
HELIX ANGLE OF THE GEAR....................	30.000 DEG
HELIX DIRECTION............................	RIGHT HAND
ADDENDUM...................................	0.167 IN
DIRECTION OF GEAR ROTATION.................	COUNTER CLOCKWISE
SURFACE STRENGTH...........................	220.000 KSI
WEIBULL EXPONENT...........................	2.500
LOAD-LIFE FACTOR...........................	8.930
THE INVOLUTE CONTACT RATIO OF THE MESH IS.	1.330
THE FACE CONTACT RATIO OF THE MESH IS.....	0.716

FORCES

AXIAL FORCE................................	240.000 LBS
RADIAL FORCE...............................	174.706 LBS
TANGENTIAL FORCE...........................	415.692 LBS
TOTAL FORCE................................	510.805 LBS
ADJUSTED DYNAMIC CAPACITY..................	3224.619 LBS

INTERMEDIATE GEAR CHARACTERISTICS AND MOUNTING

GEAR #1 IN MESH WITH PINION

NUMBER OF TEETH............................	45
PITCH RADIUS...............................	4.330 IN
NORMAL PITCH...............................	6.00 1./IN

NORMAL PRESSURE ANGLE.....................	20.00 DEG
FACE WIDTH................................	0.750 IN
HELIX ANGLE OF THE GEAR...................	30.000 DEG
HELIX DIRECTION...........................	LEFT HAND
ADDENDUM..................................	0.167 IN
DIRECTION OF GEAR ROTATION................	CLOCKWISE
SURFACE STRENGTH..........................	220.000 KSI
WEIBULL EXPONENT..........................	2.500
LOAD-LIFE FACTOR..........................	8.930

FORCES

AXIAL FORCE...............................	-240.000 LBS
RADIAL FORCE..............................	174.706 LBS
TANGENTIAL FORCE..........................	415.692 LBS
TOTAL FORCE...............................	510.805 LBS
ADJUSTED DYNAMIC CAPACITY.................	3471.650 LBS

GEAR #2 IN MESH WITH OUTPUT GEAR

NUMBER OF TEETH...........................	15
PITCH RADIUS..............................	2.069 IN
NORMAL PITCH..............................	4.00 1./IN
NORMAL PRESSURE ANGLE.....................	20.00 DEG
FACE WIDTH................................	1.250 IN
HELIX ANGLE OF THE GEAR...................	25.000 DEG
HELIX DIRECTION...........................	RIGHT HAND
ADDENDUM..................................	0.250 IN
DIRECTION OF GEAR ROTATION................	CLOCKWISE
SURFACE STRENGTH..........................	220.000 KSI
WEIBULL EXPONENT..........................	2.500
LOAD-LIFE FACTOR..........................	8.930

FORCES

AXIAL FORCE...............................	-405.714 LBS
RADIAL FORCE..............................	349.411 LBS
TANGENTIAL FORCE..........................	870.055 LBS
TOTAL FORCE...............................	1021.611 LBS
ADJUSTED DYNAMIC CAPACITY.................	7419.580 LBS

THE SECOND INVOLUTE CONTACT RATIO IS......	1.411
THE SECOND FACE CONTACT RATIO IS..........	0.673

SPEED OF INTERMEDIATE SHAFT...............	1666.67 RPM
TORQUE ON INTERMEDIATE GEARS..............	1800.000 LB-IN
INTERMEDIATE SHAFT ANGLE..................	30.00 DEG

DOUBLE OVERHUNG BEARING MOUNTING

```
GEAR#1=====BRG#1======BRG#2======GEAR#2
  <----C----->          <----E---->
  <---------------D-------------->
```

DISTANCE C................................	3.000 IN
DISTANCE D................................	11.000 IN
DISTANCE E................................	3.000 IN

BEARING #1

SINGLE ROW BALL BEARING

STATIC CAPACITY...........................	13100.000 LBS
CONTACT ANGLE.............................	25.00 DEG
BASIC DYNAMIC CAPACITY....................	4160.000 LBS
ROTATION FACTOR...........................	1.00
WEIBULL EXPONENT..........................	1.11
LIFE ADJUSTMENT FACTOR....................	6.00

AXIAL FORCE...............................	-645.714 LBS
RADIAL FORCE..............................	17.115 LBS
TANGENTIAL FORCE..........................	1062.192 LBS
TOTAL EQUIVALENT RADIAL FORCE.............	1062.330 LBS
ADJUSTED DYNAMIC CAPACITY.................	5241.271 LBS

BEARING #2

SINGLE ROW ROLLER BEARING

BASIC DYNAMIC CAPACITY....................	8810.000 LBS
ROTATION FACTOR...........................	1.00
WEIBULL EXPONENT..........................	1.13
LIFE ADJUSTMENT FACTOR....................	6.00

RADIAL FORCE..............................	691.442 LBS
TANGENTIAL FORCE..........................	1659.600 LBS
TOTAL EQUIVALENT RADIAL FORCE.............	1797.878 LBS
ADJUSTED DYNAMIC CAPACITY.................	10869.188 LBS

OUTPUT GEAR CHARACTERISTICS AND MOUNTING

OVERHUNG MOUNTING

```
GEAR=======BRG#1=======BRG#2
  <----A----->
  <----------B----------->
```

DISTANCE A................................	3.000 IN
DISTANCE B................................	8.000 IN

BEARING #1

SINGLE ROW ROLLER BEARING

BASIC DYNAMIC CAPACITY....................	12070.000 LBS
ROTATION FACTOR...........................	1.00
WEIBULL EXPONENT..........................	1.13
LIFE ADJUSTMENT FACTOR....................	6.00
RADIAL FORCE..............................	1062.671 LBS
TANGENTIAL FORCE..........................	1392.089 LBS
TOTAL EQUIVALENT RADIAL FORCE.............	1751.336 LBS
ADJUSTED DYNAMIC CAPACITY.................	20773.549 LBS

BEARING #2

SINGLE ROW BALL BEARING

STATIC CAPACITY...........................	13100.000 LBS
CONTACT ANGLE.............................	25.00 DEG
BASIC DYNAMIC CAPACITY....................	5690.000 LBS
ROTATION FACTOR...........................	1.00
WEIBULL EXPONENT..........................	1.11
LIFE ADJUSTMENT FACTOR....................	6.00
AXIAL FORCE...............................	-405.714 LBS
RADIAL FORCE..............................	-713.259 LBS
TANGENTIAL FORCE..........................	-522.033 LBS
TOTAL EQUIVALENT RADIAL FORCE.............	883.888 LBS
ADJUSTED DYNAMIC CAPACITY.................	10339.416 LBS

OUTPUT GEAR

NUMBER OF TEETH...........................	45
PITCH RADIUS..............................	6.207 IN
NORMAL PITCH..............................	4.00 1./IN
NORMAL PRESSURE ANGLE.....................	20.00 DEG
FACE WIDTH................................	1.250 IN
HELIX ANGLE OF THE GEAR...................	25.000 DEG
HELIX DIRECTION...........................	LEFT HAND
ADDENDUM..................................	0.250 IN
DIRECTION OF GEAR ROTATION................	COUNTER CLOCKWISE
SURFACE STRENGTH..........................	220.000 KSI
WEIBULL EXPONENT..........................	2.500
LOAD-LIFE FACTOR..........................	8.930

FORCES

AXIAL FORCE..............................	-405.714 LBS
RADIAL FORCE.............................	349.411 LBS
TANGENTIAL FORCE.........................	870.055 LBS
TOTAL FORCE..............................	1021.611 LBS
ADJUSTED DYNAMIC CAPACITY................	7987.975 LBS

COMPONENT AND TRANSMISSION

OUTPUT DYNAMIC CAPACITY AND LIFE

PINION		
DYNAMIC CAPACITY.......................	34089.20	LB-IN
LOAD-LIFE EXPONENT.....................	8.93	
WEIBULL EXPONENT.......................	2.50	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	0.1399562E+08	
L10 LIFE...............................	0.4198687E+09	HOURS
MEAN LIFE..............................	0.9413293E+09	HOURS
PINION BEARING #1		
DYNAMIC CAPACITY.......................	27864.66	LB-IN
LOAD-LIFE EXPONENT.....................	3.30	
WEIBULL EXPONENT.......................	1.13	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	224.7904	
L10 LIFE...............................	6743.712	HOURS
MEAN LIFE..............................	44241.79	HOURS
PINION BEARING #2		
DYNAMIC CAPACITY.......................	33978.84	LB-IN
LOAD-LIFE EXPONENT.....................	3.00	
WEIBULL EXPONENT.......................	1.11	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	249.1405	
L10 LIFE...............................	7474.215	HOURS
MEAN LIFE..............................	50267.11	HOURS
INTERMEDIATE GEAR #1 MESHING WITH PINION		
DYNAMIC CAPACITY.......................	36700.69	LB-IN
LOAD-LIFE EXPONENT.....................	8.93	
WEIBULL EXPONENT.......................	2.50	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	0.2705609E+08	
L10 LIFE...............................	0.8116827E+09	HOURS
MEAN LIFE..............................	0.1819761E+10	HOURS
INTERMEDIATE GEAR #2 MESHING WITH OUTPUT GEAR		
DYNAMIC CAPACITY.......................	39218.20	LB-IN
LOAD-LIFE EXPONENT.....................	8.93	
WEIBULL EXPONENT.......................	2.50	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	0.4892925E+08	
L10 LIFE...............................	0.1467878E+10	HOURS
MEAN LIFE..............................	0.3290924E+10	HOURS

INTERMEDIATE BEARING #1

DYNAMIC CAPACITY........................	26642.27	LB-IN
LOAD-LIFE EXPONENT......................	3.00	
WEIBULL EXPONENT........................	1.11	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	120.0970	
L10 LIFE................................	3602.910	HOURS
MEAN LIFE...............................	24231.02	HOURS

INTERMEDIATE BEARING #2

DYNAMIC CAPACITY........................	32646.06	LB-IN
LOAD-LIFE EXPONENT......................	3.30	
WEIBULL EXPONENT........................	1.13	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	379.0898	
L10 LIFE................................	11372.69	HOURS
MEAN LIFE...............................	74609.98	HOURS

OUTPUT GEAR

DYNAMIC CAPACITY........................	42222.61	LB-IN
LOAD-LIFE EXPONENT......................	8.93	
WEIBULL EXPONENT........................	2.50	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	0.9458914E+08	
L10 LIFE................................	0.2837674E+10	HOURS
MEAN LIFE...............................	0.6361955E+10	HOURS

OUTPUT GEAR BEARING #1

DYNAMIC CAPACITY........................	64052.32	LB-IN
LOAD-LIFE EXPONENT......................	3.30	
WEIBULL EXPONENT........................	1.13	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	3504.819	
L10 LIFE................................	105144.6	HOURS
MEAN LIFE...............................	689795.6	HOURS

OUTPUT GEAR BEARING #2

DYNAMIC CAPACITY........................	63167.36	LB-IN
LOAD-LIFE EXPONENT......................	3.00	
WEIBULL EXPONENT........................	1.11	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	1600.652	
L10 LIFE................................	48019.56	HOURS
MEAN LIFE...............................	322950.9	HOURS

TRANSMISSION

DYNAMIC CAPACITY........................	19752.74	LB-IN
LOAD-LIFE EXPONENT......................	3.12	
WEIBULL EXPONENT........................	1.12	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	57.11895	
L10 LIFE................................	1713.568	HOURS
MEAN LIFE...............................	11425.65	HOURS
MEAN COMPONENT LIFE.....................	9831.153	HOURS

APPENDIX D

TWO STAGE HELICAL EXAMPLE OUTPUT FILE

COMBINED OVERALL TRANSMISSION

INPUT SPEED..............................	5000.00 RPM
OUTPUT SPEED.............................	555.56 RPM
SPEED REDUCTION RATIO....................	9.00
TRANSMITTED POWER........................	47.600 HP
INPUT TORQUE.............................	600.000 LB-IN
DIRECTION OF INPUT.......................	COUNTER CLOCKWISE
OUTPUT TORQUE............................	5400.000 LB-IN
DIRECTION OF OUTPUT......................	COUNTER CLOCKWISE

SINGLE MESH HELICAL GEAR REDUCTION RESULTS

INPUT SPEED..............................	5000.00 RPM
OUTPUT SPEED.............................	1666.67 RPM
SPEED REDUCTION RATIO....................	3.00
TRANSMITTED POWER........................	47.600 HP
INPUT TORQUE.............................	600.000 LB-IN
DIRECTION OF INPUT.......................	COUNTER CLOCKWISE
OUTPUT TORQUE............................	1800.000 LB-IN
DIRECTION OF OUTPUT......................	CLOCKWISE

PINION CHARACTERISTICS AND MOUNTING

OVERHUNG MOUNTING

```
GEAR=======BRG#1=======BRG#2
  <----A----->
  <----------B----------->
```

DISTANCE A...............................	3.000 IN
DISTANCE B...............................	8.000 IN

BEARING #1

SINGLE ROW ROLLER BEARING

BASIC DYNAMIC CAPACITY...................	4070.000 LBS
ROTATION FACTOR..........................	1.00
WEIBULL EXPONENT.........................	1.13
LIFE ADJUSTMENT FACTOR...................	6.00

RADIAL FORCE..............................	210.247 LBS
TANGENTIAL FORCE..........................	665.107 LBS
TOTAL EQUIVALENT RADIAL FORCE.............	697.547 LBS
ADJUSTED DYNAMIC CAPACITY.................	3599.427 LBS

BEARING #2

SINGLE ROW BALL BEARING

STATIC CAPACITY...........................	7880.000 LBS
CONTACT ANGLE.............................	25.00 DEG
BASIC DYNAMIC CAPACITY....................	2248.000 LBS
ROTATION FACTOR...........................	1.00
WEIBULL EXPONENT..........................	1.11
LIFE ADJUSTMENT FACTOR....................	6.00
AXIAL FORCE...............................	240.000 LBS
RADIAL FORCE..............................	-35.541 LBS
TANGENTIAL FORCE..........................	-249.415 LBS
TOTAL EQUIVALENT RADIAL FORCE.............	312.093 LBS
ADJUSTED DYNAMIC CAPACITY.................	1963.809 LBS

PINION

NUMBER OF TEETH...........................	15
PITCH RADIUS..............................	1.443 IN
NORMAL PITCH..............................	6.00 1./IN
NORMAL PRESSURE ANGLE.....................	20.00 DEG
FACE WIDTH................................	0.750 IN
HELIX ANGLE OF THE GEAR...................	30.000 DEG
HELIX DIRECTION...........................	RIGHT HAND
ADDENDUM..................................	0.167 IN
DIRECTION OF GEAR ROTATION................	COUNTER CLOCKWISE
SURFACE STRENGTH..........................	220.000 KSI
WEIBULL EXPONENT..........................	2.500
LOAD-LIFE FACTOR..........................	8.930
THE INVOLUTE CONTACT RATIO OF THE MESH IS.	1.330
THE FACE CONTACT RATIO OF THE MESH IS.....	0.716

FORCES

AXIAL FORCE...............................	240.000 LBS
RADIAL FORCE..............................	174.706 LBS
TANGENTIAL FORCE..........................	415.692 LBS
TOTAL FORCE...............................	510.805 LBS
ADJUSTED DYNAMIC CAPACITY.................	3224.619 LBS

OUTPUT GEAR CHARACTERISTICS

OUTPUT GEAR

NUMBER OF TEETH..........................	45
PITCH RADIUS.............................	4.330 IN
NORMAL PITCH.............................	6.00 1./IN
NORMAL PRESSURE ANGLE....................	20.00 DEG
FACE WIDTH...............................	0.750 IN
HELIX ANGLE OF THE GEAR..................	30.000 DEG
HELIX DIRECTION..........................	LEFT HAND
ADDENDUM.................................	0.167 IN
DIRECTION OF GEAR ROTATION...............	CLOCKWISE
SURFACE STRENGTH.........................	220.000 KSI
WEIBULL EXPONENT.........................	2.500
LOAD-LIFE FACTOR.........................	8.930

FORCES

AXIAL FORCE..............................	240.000 LBS
RADIAL FORCE.............................	174.706 LBS
TANGENTIAL FORCE.........................	415.692 LBS
TOTAL FORCE..............................	510.805 LBS
ADJUSTED DYNAMIC CAPACITY................	3471.650 LBS

COMPONENT AND TRANSMISSION

OUTPUT DYNAMIC CAPACITY AND LIFE

PINION

DYNAMIC CAPACITY........................	34089.20	LB-IN
LOAD-LIFE EXPONENT......................	8.93	
WEIBULL EXPONENT........................	2.50	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	0.1399562E+08	
L10 LIFE................................	0.4198686E+09	HOURS
MEAN LIFE...............................	0.9413292E+09	HOURS

PINION BEARING #1

DYNAMIC CAPACITY........................	27864.66	LB-IN
LOAD-LIFE EXPONENT......................	3.30	
WEIBULL EXPONENT........................	1.13	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	224.7904	
L10 LIFE................................	6743.712	HOURS
MEAN LIFE...............................	44241.78	HOURS

PINION BEARING #2

DYNAMIC CAPACITY........................	33978.84	LB-IN
LOAD-LIFE EXPONENT......................	3.00	
WEIBULL EXPONENT........................	1.11	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	249.1405	
L10 LIFE................................	7474.215	HOURS
MEAN LIFE...............................	50267.11	HOURS

```
OUTPUT GEAR
     DYNAMIC CAPACITY........................ 36700.69       LB-IN
     LOAD-LIFE EXPONENT...................... 8.93
     WEIBULL EXPONENT........................ 2.50
     L10 LIFE IN MILLION OUTPUT ROTATIONS.... 0.2705609E+08
     L10 LIFE................................ 0.8116826E+09  HOURS
     MEAN LIFE............................... 0.1819761E+10  HOURS

TRANSMISSION
     DYNAMIC CAPACITY........................ 24795.19       LB-IN
     LOAD-LIFE EXPONENT...................... 3.90
     WEIBULL EXPONENT........................ 1.12
     L10 LIFE IN MILLION OUTPUT ROTATIONS.... 127.1087
     L10 LIFE................................ 3813.262       HOURS
     MEAN LIFE............................... 25309.59       HOURS
     MEAN COMPONENT LIFE..................... 23530.29       HOURS

SINGLE MESH HELICAL GEAR REDUCTION RESULTS


INPUT SPEED...............................     1666.67  RPM
OUTPUT SPEED..............................      555.56  RPM
SPEED REDUCTION RATIO.....................        3.00
TRANSMITTED POWER.........................      47.600 HP
INPUT TORQUE..............................    1800.000 LB-IN
DIRECTION OF INPUT........................   CLOCKWISE
OUTPUT TORQUE.............................    5400.000 LB-IN
DIRECTION OF OUTPUT.......................   COUNTER CLOCKWISE


PINION CHARACTERISTICS AND MOUNTING

OVERHUNG MOUNTING

          GEAR=======BRG#1=======BRG#2
            <----A----->
            <----------B----------->

DISTANCE A................................       3.000 IN
DISTANCE B................................       8.000 IN

     BEARING #1

SINGLE ROW ROLLER BEARING

BASIC DYNAMIC CAPACITY....................    8810.000 LBS
ROTATION FACTOR...........................        1.00
WEIBULL EXPONENT..........................        1.13
LIFE ADJUSTMENT FACTOR....................        6.00
```

RADIAL FORCE..............................	691.442 LBS
TANGENTIAL FORCE..........................	1659.600 LBS
TOTAL EQUIVALENT RADIAL FORCE.............	1797.878 LBS
ADJUSTED DYNAMIC CAPACITY.................	10869.188 LBS

BEARING #2

SINGLE ROW BALL BEARING

STATIC CAPACITY...........................	13100.000 LBS
CONTACT ANGLE.............................	25.00 DEG
BASIC DYNAMIC CAPACITY....................	4160.000 LBS
ROTATION FACTOR...........................	1.00
WEIBULL EXPONENT..........................	1.11
LIFE ADJUSTMENT FACTOR....................	6.00

AXIAL FORCE...............................	-645.714 LBS
RADIAL FORCE..............................	17.115 LBS
TANGENTIAL FORCE..........................	-1062.192 LBS
TOTAL EQUIVALENT RADIAL FORCE.............	1062.330 LBS
ADJUSTED DYNAMIC CAPACITY.................	5241.271 LBS

PINION

NUMBER OF TEETH...........................	15
PITCH RADIUS..............................	2.069 IN
NORMAL PITCH..............................	4.00 1./IN
NORMAL PRESSURE ANGLE.....................	20.00 DEG
FACE WIDTH................................	1.250 IN
HELIX ANGLE OF THE GEAR...................	25.000 DEG
HELIX DIRECTION...........................	RIGHT HAND
ADDENDUM..................................	0.250 IN
DIRECTION OF GEAR ROTATION................	CLOCKWISE
SURFACE STRENGTH..........................	220.000 KSI
WEIBULL EXPONENT..........................	2.500
LOAD-LIFE FACTOR..........................	8.930

THE INVOLUTE CONTACT RATIO OF THE MESH IS.	1.411
THE FACE CONTACT RATIO OF THE MESH IS.....	0.673

FORCES

AXIAL FORCE...............................	-405.714 LBS
RADIAL FORCE..............................	349.411 LBS
TANGENTIAL FORCE..........................	870.055 LBS
TOTAL FORCE...............................	1021.611 LBS
ADJUSTED DYNAMIC CAPACITY.................	7419.579 LBS

OUTPUT GEAR CHARACTERISTICS AND MOUNTING

OVERHUNG MOUNTING

```
          GEAR=======BRG#1=======BRG#2
            <----A----->
            <----------B----------->
```

```
DISTANCE A................................          3.000 IN
DISTANCE B................................          8.000 IN
```

BEARING #1

SINGLE ROW ROLLER BEARING

```
BASIC DYNAMIC CAPACITY....................      12070.000 LBS
ROTATION FACTOR...........................          1.00
WEIBULL EXPONENT..........................          1.13
LIFE ADJUSTMENT FACTOR....................          6.00

RADIAL FORCE..............................       1062.671 LBS
TANGENTIAL FORCE..........................       1392.089 LBS
TOTAL EQUIVALENT RADIAL FORCE.............       1751.337 LBS
ADJUSTED DYNAMIC CAPACITY.................      20773.549 LBS
```

BEARING #2

SINGLE ROW BALL BEARING

```
STATIC CAPACITY...........................      13100.000 LBS
CONTACT ANGLE.............................         25.00 DEG
BASIC DYNAMIC CAPACITY....................       5690.000 LBS
ROTATION FACTOR...........................          1.00
WEIBULL EXPONENT..........................          1.11
LIFE ADJUSTMENT FACTOR....................          6.00

AXIAL FORCE...............................       -405.714 LBS
RADIAL FORCE..............................       -713.259 LBS
TANGENTIAL FORCE..........................       -522.033 LBS
TOTAL EQUIVALENT RADIAL FORCE.............        883.888 LBS
ADJUSTED DYNAMIC CAPACITY.................      10339.416 LBS
```

OUTPUT GEAR

```
NUMBER OF TEETH...........................       45
PITCH RADIUS..............................          6.207 IN
NORMAL PITCH..............................          4.00  1./IN
NORMAL PRESSURE ANGLE.....................         20.00  DEG
FACE WIDTH................................          1.250 IN
HELIX ANGLE OF THE GEAR...................         25.000 DEG
HELIX DIRECTION...........................         LEFT HAND
ADDENDUM..................................          0.250  IN
```

DIRECTION OF GEAR ROTATION................	COUNTER CLOCKWISE
SURFACE STRENGTH..........................	220.000 KSI
WEIBULL EXPONENT..........................	2.500
LOAD-LIFE FACTOR..........................	8.930

FORCES

AXIAL FORCE...............................	-405.714 LBS
RADIAL FORCE..............................	349.411 LBS
TANGENTIAL FORCE..........................	870.055 LBS
TOTAL FORCE...............................	1021.611 LBS
ADJUSTED DYNAMIC CAPACITY.................	7987.975 LBS

COMPONENT AND TRANSMISSION

OUTPUT DYNAMIC CAPACITY AND LIFE

PINION

DYNAMIC CAPACITY........................	39218.19	LB-IN
LOAD-LIFE EXPONENT......................	8.93	
WEIBULL EXPONENT........................	2.50	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	0.4892922E+08	
L10 LIFE................................	0.1467877E+10	HOURS
MEAN LIFE...............................	0.3290922E+10	HOURS

PINION BEARING #1

DYNAMIC CAPACITY........................	32646.06	LB-IN
LOAD-LIFE EXPONENT......................	3.30	
WEIBULL EXPONENT........................	1.13	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	379.0897	
L10 LIFE................................	11372.69	HOURS
MEAN LIFE...............................	74609.97	HOURS

PINION BEARING #2

DYNAMIC CAPACITY........................	26642.26	LB-IN
LOAD-LIFE EXPONENT......................	3.00	
WEIBULL EXPONENT........................	1.11	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	120.0970	
L10 LIFE................................	3602.909	HOURS
MEAN LIFE...............................	24231.01	HOURS

OUTPUT GEAR

DYNAMIC CAPACITY........................	42222.61	LB-IN
LOAD-LIFE EXPONENT......................	8.93	
WEIBULL EXPONENT........................	2.50	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	0.9458908E+08	
L10 LIFE................................	0.2837672E+10	HOURS
MEAN LIFE...............................	0.6361952E+10	HOURS

OUTPUT GEAR BEARING #1

DYNAMIC CAPACITY........................	64052.32	LB-IN
LOAD-LIFE EXPONENT......................	3.30	
WEIBULL EXPONENT........................	1.13	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	3504.818	
L10 LIFE................................	105144.5	HOURS
MEAN LIFE...............................	689795.4	HOURS

OUTPUT GEAR BEARING #2

DYNAMIC CAPACITY........................	63167.36	LB-IN
LOAD-LIFE EXPONENT......................	3.00	
WEIBULL EXPONENT........................	1.11	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	1600.652	
L10 LIFE................................	48019.55	HOURS
MEAN LIFE...............................	322950.8	HOURS

TRANSMISSION

DYNAMIC CAPACITY........................	23405.64	LB-IN
LOAD-LIFE EXPONENT......................	3.08	
WEIBULL EXPONENT........................	1.11	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	91.54487	
L10 LIFE................................	2746.346	HOURS
MEAN LIFE...............................	18369.87	HOURS
MEAN COMPONENT LIFE.....................	16886.45	HOURS

COMBINED OVERALL TRANSMISSION

OUTPUT DYNAMIC CAPACITY AND LIFE
BASED ON INDIVIDUAL COMPONENT LIVES AND CAPACITIES

DYNAMIC CAPACITY........................	19752.74	LB-IN
LOAD-LIFE EXPONENT......................	3.12	
WEIBULL EXPONENT........................	1.12	
L10 LIFE IN MILLION OUTPUT ROTATIONS....	57.11906	
L10 LIFE................................	1713.572	HOURS
MEAN LIFE...............................	11425.55	HOURS
MEAN COMPONENT LIFE.....................	9831.151	HOURS

APPENDIX E

SYMBOLS

Variables

a - bearing life adjustment factor, gear addendum (mm)

A - bearing location (mm)

A_o - bevel back-cone distance (mm)

B - bearing location (mm)

$\boldsymbol{B}$ - gear surface material constant (MPa)

C - gear center distance, bearing location (mm)

$\boldsymbol{C}$ - component dynamic capacity (kN)

$\boldsymbol{C}_o$ - bearing static radial capacity (kN)

$\boldsymbol{C}_{oi}$ - component dynamic capacity for one-million output rotations (kN)

D - axial distance between two intermediate gears (mm)

$\boldsymbol{D}$ - dynamic capacity (kN-m)

e - acceptable error (mm)

E - bearing location (mm)

E_ℓ - elastic modulus (MPa)

f - gear face width (mm)

F - force (kN)

h - one-half intermediate shaft separation (mm)

ℓ - life (hours or million output rotations)

ℓ_c - load cycles per rotation

ℓ_o - life in million output rotations

m - gear module (mm)

m_a - axial contact ratio

m_N - spiral bevel load sharing ratio

n - overall gear ratio

$n1$ - first-stage gear ratio

$n2$ - second-stage gear ratio

N - number of teeth

N_g - number of teeth on gear

N_p - number of parallel load paths

p_x - axial pitch (mm)

P_d - diametral pitch (in^{-1})

R - gear radius (mm)

$\boldsymbol{R}$ - reliability

S_{ac} - surface fatigue strength for one-million load cycles (MPa)

ST - output gear sign switch (+ for external / - for internal)

T - torque (kN-m)

V - bearing load adjustment factor

W - input gear load (kN)

z_o - stress depth (mm)

Z - length of line of action (mm)

β - transmission coupling angle (degrees)

Γ - spiral bevel gear cone angle (degrees)

Γ - gamma function

η - vectorized length of contact zone (mm)

η_I - vectorized distance to point of maximum contact stress (mm)

$\delta\theta$ - addendum angle (radians)

θ - characteristic life (hours)

Λ - dual-bevel input-shaft separation angle (degrees)

ν - Poisson's ratio

ρ - radius of curvature (mm)

Σ - shaft angle (degrees)

$\sum$ - mathematical summation

τ_o - contact shear stress (MPa)

ϕ_n - normal pressure angle (degrees)

ψ - helix or spiral angle (degrees)

ω - rotational speed (RPM)

<u>Subscripts</u>

a - axial

b - base

ae - equivalent axial

av - average

e - equivalent

eff - effective

g - gear

ii - meshing with input gear

io - meshing with output gear

j - index

n - normal

nr - normal pressure angle on ring

ns - normal pressure angle on sun

o - output

p - planet

pr - planet meshing with ring

ps - planet meshing with sun

r - radial

ra - radial on first bearing

rb - radial on second bearing

re - equivalent radial

ri - input radial

ro - output radial

s - sun or system

t - tangential or tooth

ta - tangential on first bearing

tb - tangential on second bearing

te - equivalent tangential

ti - input tangential

to - output tangential

tr - tangential on ring

ts - tangential on sun

1 - first

2 - second

10 - ninety-percent reliability

Superscripts

b - Weibull slope

bi - component Weibull slope

bs - system Weibull slope

c - exponent of proportionality

h - exponent of proportionality

p - load-life factor

pi - component load-life factor

p - system load-life factor

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13. ABSTRACT ***(Maximum 200 words)***

This report describes a computer program, "TLIFE," which models the service life of a transmission. The program is written in ANSI standard Fortran 77 and has an executable size of about 157 K bytes for use on a personal computer running DOS. It can also be compiled and executed in UNIX. The computer program can analyze any one of eleven unit transmissions either singly or in a series combination of up to twenty-five unit transmissions. Metric or English unit calculations are performed with the same routines using consistent input data and a units flag. Primary outputs are: the dynamic capacity of the transmission and the mean lives of the transmission and of the sum of its components. The program uses a modular approach to separate the load analyses from the system life calculations. The program and its input and output data files are described herein. Three examples illustrate its use. A development of the theory behind the analysis in the program is included after the examples.

14. SUBJECT TERMS
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